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RESEARCH MEMORANDUM

INVESTIGATION OF WATER-SPRAY COOLING OF TURBINE BLADES IN A TURBOJET ENGINE

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SUMMARY

An analytical and experimental investigation of water-spray cooling of turbine blades was conducted with a representative centrifugal-flow engine (J33-A-9). The analytical part of the investigation provided an indication of the gains in thrust available with spray cooling. This analysis was conducted for a range of inlet-gas temperatures and engine speeds above the rated condition (inlet-gas temperature of 1670° F and engine speed of 11,500 rpm). With the assumption of adequate spray cooling at a coolant-to-gas flow ratio of 3 percent, calculations for the sea-level static condition indicated that a thrust increase of 40 percent over rated thrust may be realized by engine operation at an inlet-gas temperature of 2000° F and an overspeed of 10 percent. For these engine operating conditions, the analysis further indicated that a combination of compressor water injection and spray cooling will provide a thrust increase of 52 percent with a total coolant-to-gas flow ratio of 7 percent.

The experimental phase of the investigation was conducted with several water-injection configurations, and data were obtained over a range of engine speeds. Water-spray cooling did not appear satisfactory for temporarily augmenting cooling at the leading and trailing edges of air-cooled blades with the type of water-injection configurations used in this investigation.

Water-spray cooling for standard, solid, Haynes-Stellite 21 blades resulted in an average midspan blade temperature of about 500° F, as compared with an uncooled-blade temperature of 1260° F, at a coolant-to-gas flow ratio of 0.0089 (2360 lb/hr at rated speed of 11,500 rpm) with inner-diaphragm nozzles, the best configuration investigated. A temperature difference of about 600° F between the blade leading and trailing edge occurred at this operating condition. No appreciable radial temperature difference was observed at the rated condition. Two blade failures, one an instrumented blade with considerable blade metal removed, occurred during rated-speed operation, probably as a result of large temperature differences along the blade chord. Calculations based on strength properties of the blade material and extrapolation of rated-speed spray-cooling data indicated that a cooling-spray rate of 20 gallons per minute was required for engine operation at an inlet-gas temperature of 2000° F and an overspeed condition of 10 percent.

INTRODUCTION

A liquid-cooling process in which the latent heat of vaporization of water is utilized has excellent turbine-cooling possibilities. One method of utilizing the latent heat of vaporization is direct injection of water sprays into the gas stream thereby permitting impingement of the water sprays on the heated rotor-blade surfaces. Liquid boiling occurs upon contact, thus effecting high heat-transfer rates. Since the coolant is dissipated in the gas stream, the coolant supply must be constantly replenished in order to provide continuous cooling. Stationary and possibly marine installations, when a large water supply can be provided and minimum weight requirements are not so critical, lend themselves readily to a continuous spray-cooling application. In aircraft turbine-engine installations in which an inexhaustable water supply cannot be provided and minimum weight is of primary importance, spray cooling is necessarily limited to short-time application. If large thrust increases can be realized by use of spray cooling, however, even short-time application would be desirable to assist in sudden acceleration, take-off, climb, or combat maneuvers.

Water-spray cooling has been investigated with a static cascade and a small-scale turbosupercharger (ref. 1). The investigation was designed to determine the feasibility of applying spray cooling to turbines fabricated from noncritical materials for marine applications. An adjustable spray bar was provided in a passage between adjacent stator blades which permitted water injection in several directions, upstream, downstream, and cross channel with respect to the gas flow. The cooling effectiveness achieved was not noticeably affected by the direction of water injection. Substantial blade temperature reductions were reported in both the static cascade and the turbosupercharger investigations. Because of the small blade size of the turbosupercharger, a single thermocouple was provided at the blade midchord position and complete temperature distributions around the blade could not be obtained.

Application of spray cooling to a full-scale turbojet engine was investigated in England (refs. 2 to 4), and substantial blade temperature reductions were also reported. In these investigations the water spray was directed downstream from the trailing edge of the stator blades, and a range of engine speeds and coolant-to-gas flow rates was considered. For the most severe operating conditions encountered, at design speed and an effective gas temperature of 1400° F, a temperature reduction of 613° F in the midchord (1/3-span position) temperature was observed with a coolant-to-gas flow ratio of approximately 0.0082. Although appreciable blade temperature gradients were reported, the results appeared sufficiently promising to warrant further experimental investigation of spray cooling.

Because of its apparent effectiveness, spray cooling was considered as a means of reducing the leading- and trailing-edge temperatures of internally air-cooled blades as well as standard solid blades of turbojet engines. Turbine-cooling investigations conducted at the NACA Lewis laboratory with internally air-cooled blades demonstrated an unequal chordwise blade temperature distribution with high leading- and trailing-edge temperatures (refs. 5 and 6). The midchord temperatures, however, were believed to be sufficiently low to permit turbine operation at higher gas temperature levels if the leading- and trailing-edge temperatures could be reduced.

With the assumption that spray cooling is sufficiently effective to permit engine operation at elevated speeds and gas temperatures, the analysis reported herein was made to determine the thrust increase obtainable with a J33-A-9 engine using water-spray cooling and compressor water injection over a range of inlet-gas temperatures and engine speeds above rated engine conditions (inlet-gas temperature of 1670° F and engine speed of 11,500 rpm). These calculations justified further consideration of spray cooling, and the experimental phase of the water-spray cooling investigation reported herein was initiated at the NACA Lewis laboratory with a modified J33 turbojet engine.

The engine was equipped with four internally air-cooled Inconel blades (similar in profile and fabrication to blades 1 and 2 of ref. 5); 50, standard, solid, Haynes-Stellite 21 blades; and several water-injection configurations. It was believed that the water entrance angle as well as the water evaporation rate might affect the blade coverage. Consequently three types of water-injection configurations were provided in order to vary this angle and the evaporation rate. Because of their location, high water-spray velocities may be obtained with two of these configurations by utilizing the high-velocity gas stream; also, high evaporation losses may occur as a result of their location. The third type of water-injection configuration, which was similar to that employed in the British investigation, and which depended primarily upon the limited pressure drop available in this installation across the injection nozzle for water-spray acceleration, may provide lower water velocities but probably smaller evaporative losses. The water-injection configurations were investigated individually and in combination at engine speeds of 4000, 8000, and 11,500 rpm. Internally air-cooled blades were investigated over a liquid coolant-to-gas flow range from 0.001 to 0.017 (64 to 1690 lb/hr) and a constant air coolant-to-gas flow ratio of 0.05 at engine speeds of 4000 and 8000 rpm. Solid Haynes-Stellite 21 blades were investigated over a range of liquid coolant-to-gas flow ratios from 0.001 to 0.0217 (64 to 5650 lb/hr) and engine speeds of 4000, 8000, and 11,500 rpm.

APPARATUS

In order to conduct an investigation of spray cooling on a full-scale aircraft turbine engine, necessary modifications were provided on a J33-A-9 turbojet engine which is rated at 11,500 rpm, a turbine inlet-gas temperature of approximately 1670° F, and an air flow of approximately 76 pounds per second. Since the effectiveness of spray cooling was to be investigated with internally air-cooled blades as well as standard solid blades, a cooling-air supply system in addition to a water-injection system was provided.

Engine Modifications

Cooling-air supply system. - An independent cooling-air supply system similar to that employed in an experimental investigation of air-cooled turbine blades in a turbojet engine (ref. 7) was utilized. Figure 1 shows the engine tail-cone modifications. The cooling-air supply system which was employed because of the four internally air-cooled blades being investigated is described in detail in reference 7. Briefly, blade cooling air was introduced through the engine tail cone to a tube concentric with the center line of the turbine rotor. From this tube air was delivered to the blades through four radial tubes welded to the rotor face. These tubes supplied air to the blades through holes drilled from the downstream rotor face to the bottom of serrated grooves holding the blades. The two inlet tubes through which cooling air entered the tail cone were enclosed by concentric tubes through which scavenge air was passed to minimize the temperature rise of the blade cooling air.

Air-cooled blades. - Air-cooled blades similar to the one shown in figure 2(a) were investigated. Four of these blades, displaced approximately 90° from each other, and fifty, standard, 4-inch-span, Haynes-Stellite 21 blades (figs. 2(b) and 2(c)) were provided in the rotor assembly. The air-cooled blades were fabricated in a manner similar to that described in reference 5 and consisted of untwisted, formed, Inconel X shells with tapered walls. The blade shell was filled with tubes extending from tip to base. The tubes were brazed to each other and to the inner surface of the blade shell to increase the internal heat-transfer surface area. The blade shell extended through the cast hollow SAE 4130 blade base and was Nicrobrazed to the base.

Water-injection configurations. - Three types of water-injection configuration shown in figure 3 were provided. From the trailing-edge type of injection configuration, water was discharged into the gas stream through orifices provided in the stator-blade trailing edge. A 7/32-inch-diameter hole was drilled through the central portion of the stator blade from the base to within 3/8 inch from the tip. Three holes $1\frac{1}{8}$-inches apart were drilled from the trailing edge to intersect the central passage, and 3/16-inch-diameter tubes were welded into these passages. The ends of the tubes were welded shut and the desired orifice size drilled into the tube ends. Six stator blades were modified in this manner. On four blades so modified and spaced 90° apart around the stator ring circumference, the orifice tubes were $1\frac{1}{8}$-inches apart, with the lower tube 5/8 inch from the blade base. The other two blades spaced 180° apart were modified in a similar manner except that the lower orifice tube was $1\frac{1}{8}$ inches from the blade base. This was done to provide more complete spanwise injection coverage, and it was expected that such an installation would provide data as to the comparative effectiveness of various spanwise injection stations. A 0.021-inch-diameter orifice size was provided for the runs made at 4000 and 8000 rpm. This diameter was increased to 0.034 inch for operation at 11,500 rpm in order to achieve the desired range of coolant-to-gas flow ratios.

From the suction-surface type of injection configuration, water was discharged into the gas stream through orifices provided on the suction surfaces of the stator blades. A 7/32-inch-diameter hole was drilled through the central section of the stator blade from the base to within 3/8 inch from the tip. The required orifice size was achieved by drilling three holes of desired diameter from the blade suction surface to intersect the central passage. Four stator blades spaced 90° apart were modified in this manner. On two of these blades the orifices were $1\frac{1}{8}$-inches apart with the lower orifice $1\frac{1}{8}$ inches from the blade base. The other two blades were modified similarly except that the lower orifice was 5/8 inch from the blade base. A 0.021-inch orifice size was provided for the runs made at 4000 and 8000 rpm. This diameter was increased to 0.041 inch for rated-speed operation.

The third or inner-diaphragm type of injection configuration merely consisted of a hole drilled through the inner ring of the stator diaphragm between adjacent stator blades. Two such configurations were provided 180° apart. The hole diameters were changed from 0.052 inch after operation at 4000 and 8000 rpm to 0.099 inch for operation at 11,500 rpm.

Water supply system. - City water was utilized as the cooling-spray medium. A three-stage constant-displacement-type pump provided with a suitable by-pass arrangement raised the water pressure to 500 pounds per square inch. Pressure regulators permitted selection of water-injection pressures. Individual water lines were provided from the stator injection configurations on the engine to a manifold downstream of the pump. Manually operated needle valves on each line permitted operation with the desired injection configuration.

Instrumentation

The engine instrumentation and the instrumentation required for cooling-air flow measurements used in this investigation are described in detail in reference 7.

Water-coolant measurements. - Water flow rates were measured by means of a calibrated rotameter. Bourdon gages were employed to determine the water pressure. Water supply temperature was measured by means of a thermocouple located within the manifold which supplied the water lines leading to the individual injection configurations.

Blade instrumentation. - A total of 14 chromel-alumel thermocouples (36-gage wire) were installed on four air-cooled blades at the locations shown in figure 2(a). Thermocouples were also placed in the cooling-air inlet passages of two blades near the blade root as described in reference 7. Six thermocouples were provided to obtain the solid blade temperatures at the locations shown in figure 2(b). The air-cooled blades were replaced by solid blades after engine operation at 4000 and 8000 rpm. A total of 19 thermocouples were subsequently installed on six solid blades at the locations shown in figure 2(c) to obtain the solid-blade temperature distribution during rated-speed operation. A slip-ring type of thermocouple pickup similar to that described in reference 7 was employed to achieve transition from the rotating thermocouples to the stationary measuring potentiometer.

EXPERIMENTAL PROCEDURE

The various water-injection configurations were investigated over a range of operating conditions. The complete range of operating conditions covered is presented in table I. For each injection configuration or combination of configurations investigated, the engine speed was maintained constant while the water-coolant flow rate was varied from the maximum value permitted by the installation to a minimum value as indicated in table I. A complete set of data was also taken at zero water flow for each injection configuration investigated. The same procedure was followed at three engine speeds, 4000, 8000, and 11,500 rpm. Water sprays were turned on simultaneously with the engine starts to minimize thermal shock conditions in the blades throughout the investigation. The engine was shut down for blade inspection after a range of water flows from maximum to zero flow had been investigated with a particular injection configuration. The water flow rate was varied by manually adjusting the water supply pressure. During operation in which air-cooled blades as well as solid blades were being investigated, cooling-air weight flow was set by manually operated valves in the air supply line to maintain a representative cooling-air flow equivalent to a total cooling-air-to-gas flow ratio of 5 percent. The adjustable exhaust nozzle was maintained at the fully open position during engine operation at all speeds except for several runs at the rated-speed condition. The exhaust-nozzle position was altered at rated speed to obtain a variation in inlet-gas temperature, and operation was conducted over a limited range of water-coolant flow rates at each of these gas temperatures with an injection configuration consisting of a combination of stator trailing-edge and inner-diaphragm spray nozzles.

METHODS OF CALCULATION

It was assumed that thrust increases could be obtained with spray cooling by operation at engine conditions above rated. Calculations were made to determine the magnitude of those increases and thereby establish the extent of the usefulness of spray cooling.

Engine Thrust Calculations

Engine thrust calculations were made with the J33 engine for the case of spray cooling and compressor water injection for step increases in turbine inlet-gas temperature from 1670° to 2000° F at various engine speeds, from 11,500 rpm (rated engine speed) to 10 percent overspeed. The compressor characteristics employed in the calculations were obtained from reference 8 and are shown in figure 4. The lines of constant corrected speed are somewhat idealized in that they consist of straight segments, whereas the experimental constant-speed lines would be slightly curved. The following major simplifying assumptions were made: (1) The specific compressor work is constant for a given speed and compressor water-injection rate, (2) turbine efficiency is constant and the turbine stator is choked over the entire range considered, and (3) the exhaust-nozzle area may be varied.

Spray cooling. - The following general method of thrust calculation was employed: Temperature and pressure conditions at the exhaust nozzle necessary for thrust determination were obtained as a result of a position-to-position calculation of temperatures and pressures throughout the engine. Standard sea-level static conditions were assumed at the compressor inlet. Specific work was calculated from compressor-outlet conditions for an engine operating line determined by means of curves from reference 8. A 5-percent pressure drop was assumed to occur through the burners. Stator-inlet total temperature was considered to be the independent variable. With a Mach number of 1 assumed at the stator throat, conditions of static temperature and pressure at this point were obtained from the following isentropic expressions:

$$T'/T = 1 + \left(\frac{\gamma_g - 1}{2}\right)M^2 \tag{1}$$

$$p'/p = \left[1 + \left(\frac{\gamma_g - 1}{2}\right)M^2\right]^{\frac{\gamma_g}{\gamma_g - 1}} \tag{2}$$

All symbols are defined in the appendix.

The constant-speed lines given in figure 4 were then used to calculate either the compressor pressure ratio or the compressor weight flow as the turbine inlet-gas temperature was varied for a given speed. Along the horizontal portion of a constant-speed line, the pressure ratio was read from figure 4, and the compressor weight flow was calculated by means of the effective stator throat area given in reference 8 with the continuity relation. Reduction in turbine inlet-gas temperature permits a higher mass flow through the stator throat (specific volume of gas is reduced), and the point of equilibrium operation moves to the right along the horizontal portion of the constant-speed line. When the vertical portion of the constant-speed line was reached and the turbine inlet-gas temperature further decreased, the weight flow remained constant and the compressor pressure ratio was calculated for equilibrium operation of the compressor and turbine. Actual and ideal total temperatures at the turbine-rotor outlet were determined by equating specific compressor and turbine work. These quantities can be expressed in several forms which are related to each other in the following manner:

$$(\Delta h'_{1-2})_{ac} = c_{p,c}(T'_2 - T'_1) = c_{p,t}(T'_3 - T'_4)(1 + f/a)$$

$$= c_{p,t}(T'_3 - T'_{4,s})(\eta_t)(1 + f/a) \tag{3}$$

Total pressure at the turbine-rotor outlet was then calculated by isentropic relations from the ideal total temperature. This pressure was assumed to exist at the exhaust nozzle since losses in the tail cone are assumed to be negligible. The total temperature at the exhaust nozzle was also considered identical with the actual temperature at the turbine-rotor outlet except for the temperature reduction effected by evaporation of the water spray. By neglecting the heat extracted from the gas to bring the water to saturation temperature and superheating the steam, this temperature reduction is expressed as

$$\Delta T = \frac{w_w h_{fg}}{w_g c_{p,g,4}} \tag{4}$$

Since the exhaust nozzle was of the convergent type, thrust was calculated by the following expression:

$$F = \frac{w_g V_5}{g} + A_5(p_5 - p_0) \tag{5}$$

where

$$V_5 = \sqrt{\gamma_{g,5} g R T_5 M_5^2}$$

and

$$w_g = w_a + w_f + w_w$$

and A_5 equals the area required to pass the total mass flow. For nozzle pressure ratios less than or equal to the critical pressure ratio, p_0 equals p_5. In this case M_5 was computed from equation (2) and, then, T_5 from equation (1). For supercritical pressure ratios, M_5 was set equal to unity and p_5 and T_5 were calculated from equations (1) and (2).

Compressor water injection. - The effect of injecting water into the inlet of the compressor is to change the compressor characteristics. The pressure ratio and mass flow are increased, and the efficiency is usually reduced. Therefore, before calculating thrust, the new compressor characteristics were determined for various amounts of water injected. Reference 9 presents the change in characteristics with water-air ratio. The basic pressure ratio, mass flow, and efficiency of the compressor of reference 9 are slightly different from those used in the calculations of this report. Consequently, the change in compressor characteristics effected by water injection for the engine of the present investigation was taken to be in the same ratio as the change in characteristics presented in reference 9. Except for the use of experimental data from reference 9, the calculation method for the water-air compression is similar to that used in reference 10, but for the purpose of completeness the exact procedure is presented as follows: With water injected into the compressor, the compressive process was considered to consist of two phases: the first, saturated compression; and the second, adiabatic compression of the resulting water-air mixture. With the incoming air assumed at standard sea-level conditions of pressure and temperature and at a relative humidity of 50 percent, the enthalpy of the air at compressor inlet was determined by use of a psychrometric chart. By means of the enthalpy-entropy chart of reference 10, the pressure, temperature, and enthalpy at the end of the saturation compression for any given water-air ratio and for constant entropy were obtained. With the compressor-outlet pressure known from the compressor characteristics, the ideal outlet temperature was determined as follows:

$$T'_{2,s} = T'_{1a,s}\left(\frac{p'_2}{p'_{1a}}\right)^{\frac{\gamma_{a,1a}-1}{\gamma_{a,1a}}} \tag{6}$$

The ideal work of the adiabatic compressive process was then determined from

$$(\Delta h'_{1a-2})_s = c_{p,a,1a}(T'_{2,s} - T'_{1a,s}) \tag{7}$$

The total ideal work of the compressor is the sum of the work of each phase.

$$(\Delta h'_{1-2})_s = (\Delta h'_{1-1a})_s + (\Delta h'_{1a-2})_s \tag{8}$$

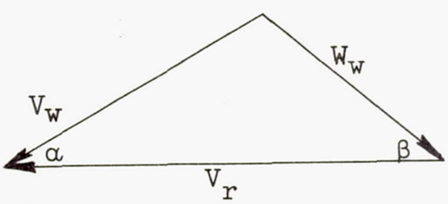
For the case of compression of a saturated mixture of water vapor and air over the total compressor pressure ratio, the second term of equation (8) is zero. The water-air ratio for this case was assumed to be that obtained from the enthalpy-entropy chart of reference 10 for isentropic compression of a saturated mixture from compressor-inlet pressure to compressor-outlet pressure. With the efficiency established by the compressor characteristics, the actual work was found from the following equation:

$$(\Delta h'_{1-2})_{ac} = \frac{(\Delta h'_{1-2})_s}{\eta_c} \tag{9}$$

When the actual work obtained from equation (9) was substituted into equation (3), thrust was calculated by the same procedure as that described previously for spray cooling except that equation (4) was omitted.

Determination of Water Entrance Angle Relative to Plane of Rotor Rotation

It may be assumed that the angle of entrance of the water spray to the rotor blades affects the manner in which the spray is distributed over the blade surface. An attempt was therefore made to calculate the water entrance angle relative to the plane of rotor rotation. The absolute and relative velocities of the water entering the rotor can be represented by the following velocity diagram:



If the absolute flow angle α (measured from the tangential direction) is known, the relative flow angle β (measured from the tangential direction) can be computed for a range of water velocities from the equation

$$\tan \beta = \frac{V_w \sin \alpha}{V_r - V_w \cos \alpha} \tag{10}$$

where

$$V_r = \frac{2\pi rN}{60}$$

These equations were solved for a range of water velocities and engine speeds of 4000, 8000, and 11,500 rpm for the blade midspan position ($r = 0.9167$ foot) for the condition that the water entered the rotor at an absolute angle equal to the stator-outlet angle of 29°. The stator trailing-edge nozzles were specifically designed to provide water injection in this direction. For the other injection configurations, however, this assumption of flow angle is approximately true only in the case that the water is sufficiently accelerated by the gases so that the water and gas flow directions are identical.

Derivation of Expression for Efficiency of Spray-Cooling Process

In order to provide a method of obtaining an indication of the coolant flow requirements for engine operation at conditions above rated speed, a means of expressing the effectiveness of the spray-cooling process is necessary. For lack of a better term, this has been designated as an efficiency. For any injection configuration considered, the efficiency of the spray-cooling process may be defined as the ratio of heat extracted from the blades to the total heat required to evaporate the coolant water. The efficiency may be expressed in equation form as follows:

$$\sigma = \frac{Q_{w,ac}}{Q_{w,t}} \tag{11}$$

If the spray-cooling process is considered in a nonrigorous manner, it may be assumed that the heat removed from the blade is equal to the heat input to the blade by convection from the gas, with the temperature drop through the water film around the blade neglected; and the following equations may be written:

$$Q_{w,ac} = HS(T_{g,e} - T_{B,av})$$

and

$$Q_{w,T} = w_w h_{fg}$$

For a rotor-blade profile the gas-to-blade heat-transfer coefficient can be expressed in terms of dimensionless numbers (ref. 11) as

$$Nu/Pr^{0.33} = \overline{F}Re^{Z} \tag{12}$$

By means of the methods of reference 11, the exponent Z and the coefficient $\overline{F}$ were evaluated for the J33 engine blade profile and found to be 0.75 and 0.06, respectively. In reference 12, the fluid-property terms which appear in these dimensionless numbers are expressed as functions of temperature. In the present report the fluid properties were based on the average temperature around the blade midspan and expressed by the following equations:

$$\mu = K_1 T_{B,av}^{0.70}$$

$$c_p = K_2 T_{B,av}^{0.19}$$

$$k = K_3 T_{B,av}^{0.85}$$

where K_1, K_2, and K_3 are proportionality constants. The exponents in the foregoing relations are the same as those used in reference 12. Writing equation (12) in terms of the gas temperature and blade temperature, combining constants, including water and gas flow terms, and incorporating it in equation (11) yield the efficiency

$$\sigma = \frac{\dfrac{T_{g,e}^{0.75}}{T_{B,av}^{0.425}}\left(T_{g,e} - T_{B,av}\right)}{\dfrac{w_w}{w_g^{0.75}}} K_4 \qquad (13)$$

where K_4 was found to be 1.024×10^{-6} and the temperatures in the ratio $T_{g,e}^{0.75}/T_{B,av}^{0.425}$ must be in $^{\circ}R$.

Application of Expression for Efficiency of Spray-Cooling Process

Application to rated engine conditions. - The expression for the efficiency of the spray-cooling process was applied to data obtained from a series of runs made at several gas temperatures and rated engine speed with an injection configuration consisting of a combination of stator trailing-edge and inner-diaphragm spray nozzles. Equation (13) was evaluated for each condition of gas temperature and coolant flow investigated at rated speed with this water-injection configuration. The average blade temperature was taken as an integrated average around the blade midspan. Effective gas temperature was considered to be the integrated average blade temperature around the midspan for operation without spray cooling.

Application to engine conditions above rated. - The expression for efficiency was also applied to estimate the coolant flow rate required for engine operation at temperatures and rotative speeds above the rated condition. Blade centrifugal stress at the midspan section was calculated for the engine speed under consideration. The centrifugal stress was multiplied by a stress-ratio factor of 3.25 to estimate the maximum allowable blade stress. The stress-ratio factor may be considered as a safety factor and is defined in reference 5. A high stress-ratio factor of 3.25 was arbitrarily selected as a possible means of accounting for large thermal gradients introduced in the blades by spray cooling. Stress-to-rupture and yield-strength data for Haynes-Stellite 21 alloy and the maximum allowable blade stress, obtained as indicated above, were used to determine the allowable blade temperature. Substitution of the allowable blade temperature and the effective gas temperature equivalent to the desired inlet-gas temperature into equation (13) and use of the relation between efficiency, coolant flow, and gas flow, as determined from data, provided the required coolant-to-gas flow ratio through an iterative process. The effective gas temperature was calculated from the turbine-inlet conditions by the procedure of reference 7, which utilizes the relation

$$\Lambda = \frac{T_{g,e} - T_{g,3}}{T''_{g,3} - T_{g,3}} = \frac{T_{g,e} - T_{g,3}}{\dfrac{(w_{g,3})^2}{2Jgc_{p,g}}} \tag{14}$$

where Λ is taken as 0.85. Although data indicate that the efficiency of the spray-cooling process increases with increasing gas temperature, the value of efficiency obtained at 1260° F and rated speed was used for all inlet-gas temperatures considered from 1670° to 2000° F. This introduces an additional safety factor into the calculations.

RESULTS AND DISCUSSION

Results of both the analytical and experimental phase of this investigation are presented and discussed. The results of each phase are presented separately and, where possible, the analytical assumptions are considered in the light of experimental data.

An analysis was made to determine the effect on thrust of increased engine speed and gas temperature for a J33-A-9 turbojet engine utilizing spray cooling and compressor water injection. Calculations were also made to determine the effect of water velocity on the water-spray entrance angle to the rotor blades.

Spray cooling. - The variation of calculated thrust at sea-level static conditions with turbine inlet-gas temperature is shown in figure 5(a). The thrust increase above rated thrust is plotted for several engine speeds up to 10 percent overspeed and for a coolant-to-gas flow ratio of zero and 3 percent. A 3-percent dilution was chosen as being sufficiently large to indicate any significant changes in thrust over the uncooled condition, and as being realistic from a spray-cooling standpoint. The uncooled-engine operating limits determined from blade material properties are indicated by vertical lines on the figure, and lines of constant percent of design turbine-exit axial critical velocity ratio and compressor surge are also shown. The dotted portion of the curves represents an extrapolation from the calculated points to the compressor surge operating limit. The figure presents the thrust increases available if it were possible to operate the engine at inlet-gas temperatures and speeds above the rated condition without cooling. The thrust increases are about one percent less than for the 3-percent-dilution spray-cooled case, indicating that the amount of thrust increase is almost exclusively due to engine operation at higher energy levels. The usefulness of spray cooling is to permit engine operation at increased gas temperatures in the range to the right of the vertical stress-limit lines, thereby achieving the large thrust increases shown provided limiting loading of the turbine does not occur or the turbine efficiency is not appreciably diminished. The lines of constant percent-of-design axial critical velocity ratio on figure 5(a) (computed for a design velocity ratio of 0.7) indicate the range of operation permitted by the turbine. For the J33 turbine a condition of maximum turbine work probably occurs for an exit axial critical velocity ratio approximately 10 percent higher than the design value. With the uncooled engine a maximum thrust increase of approximately 6 percent can be achieved for a 10-percent increase in the critical velocity ratio by increasing the engine speed and decreasing the turbine inlet-gas temperature (achieved by increasing the exhaust-nozzle opening) to prevent overstressing the blades. If spray cooling with a dilution of 3 percent is adequate to permit engine operation at a condition of 10 percent overspeed and a turbine inlet-gas temperature of 2000° F, a thrust increase of 40 percent over the rated condition is possible with only a slight increase in the turbine-exit axial critical velocity ratio. This operating condition is cited as probably representing the maximum obtainable without requiring major mechanical modifications to the engine.

In summation these calculations indicate that large gains in thrust are available if the engine can be operated at conditions of temperature and speed above rated. These thrust increases are almost solely due to operation at higher energy levels. The increase in thrust due to the addition of a water spray (3-percent dilution) is virtually negligible.

Compressor water injection. - The variation of calculated engine thrust at sea-level static conditions with turbine inlet-gas temperature for several compressor water-injection rates up to the saturation limit and engine speeds up to 10 percent overspeed is shown in figure 5(b).

As in the previous figure, the uncooled-engine operating limits are indicated by vertical lines. Lines of constant percent-of-design turbine-exit axial critical velocity ratio and compressor surge are also shown. For uncooled operation and with compressor water injection, a thrust increase of approximately 6 percent over the rated condition can be achieved if the turbine-exit axial critical velocity ratio is restricted to approximately a 10-percent increase. As indicated in the previous section, substantial thrust increases are available by engine operation at increased energy levels; however, even greater gains may be achieved by use of compressor water injection at these elevated engine operating conditions over the range where the operation is not limited by the turbine. At the maximum water-injection rate these increases are of the order of 10 percentage points above the zero water-injection condition over most of the temperature range and all the overspeed conditions considered. For 10 percent overspeed at a turbine inlet-gas temperature of 2000° F, this additional thrust increase of 10 percentage points with water injection is accomplished with approximately a 20-percent increase in the turbine-exit velocity ratio.

In figure 6 is shown the comparison of the thrust increase obtainable at a gas temperature of 2000° F with simple spray cooling, with a combination of spray cooling and compressor water injection, and without either method, if it is possible from a stress and aerodynamic design standpoint to operate the engine under all these conditions. From a stress standpoint, operation at 2000° F without adequate spray cooling would, of course, necessitate much stronger high-temperature materials than are currently available. The percentage increase in thrust over the rated value is plotted against engine speed for constant turbine inlet-gas temperature. At a turbine inlet-gas temperature of 2000° F, the maximum compressor water-injection rate (0.0403) plus a coolant-to-gas flow ratio of 3 percent for spray cooling has been selected for comparison with the zero water-injection rate and the 3-percent-dilution spray-cooled condition. The curve showing the combination of compressor water injection and spray cooling was obtained by adding the thrust increase between the zero and 3-percent spray-cooled conditions to the thrust increase obtained by compressor water injection at saturation flow at 2000° F (fig. 5(b)). A minimum thrust increase with compressor water injection of 9 percentage points and a maximum increase of 13 percentage points occur at rated and 10 percent overspeed, respectively, over the zero water-injection rate. It is therefore apparent that the greatest gains may be realized if compressor water injection can be utilized at elevated engine operating conditions. The simple spray-cooled condition for a 3-percent dilution provides only slightly greater thrust (1 percentage point) than the uncooled condition or the zero water-injection rate. It should again be noted that engine operation in the whole over-temperature, overspeed range is not possible without spray cooling using current turbine materials. With the assumption that adequate spray cooling can be achieved with a 3-percent dilution, a combination of spray cooling and compressor water injection, requiring a total dilution of 7 percent, will permit a 52-percent increase in thrust. Translated in terms of water requirements for 1-minute operation, which should be adequate for take-off application in an aircraft installation, this dilution is equivalent to 45 gallons.

The compressor water-injection calculations indicate generally that thrust increases about 10 percentage points above the zero water-injection rate can be achieved over most of the temperature range and all the overspeed conditions considered. This contrasts with a thrust increase of only 1 percentage point calculated as being obtainable with the simple spray-cooled condition in comparison with a zero water-spray rate.

The results of this analysis are, of course, affected by the simplifying assumptions cited previously. The analysis was made for a centrifugal-flow engine, and the trend of the results is approximately applicable to an axial-flow engine as well. These assumptions served the purpose of simplifying the calculations, and their net effect upon the final calculated results appears to be negligible.

Water-spray entrance angle. - Calculations were made to determine the effect of water velocity on the angle of incidence of the water relative to the plane of turbine rotation. The results of these calculations are shown in figure 7. The angle of the water spray relative to the plane of rotor rotation is plotted against the water velocity for engine speeds of 4000, 8000, and 11,500 rpm. The curves for each speed were determined for the blade midspan position. The figure indicates that as the velocity of the water leaving the stator increases, the angle of the water spray relative to the plane of turbine-rotor rotation increases, and the water strikes the blade more nearly at the aerodynamic gas entrance angle. For engine speeds of 4000, 8000, and 11,500 rpm, water velocities of 700, 1400, and 2000 feet per second, respectively, are required to provide the aerodynamic gas entrance angle of 123° measured from the plane of rotor rotation. Water velocities of this magnitude cannot be achieved simply by increasing the pressure drop across the injection nozzles. Such pressure-drop requirements would be prohibitive (a pressure drop of 26,500 lb/sq in. is theoretically required to provide a water velocity of 2000 ft/sec, whereas a maximum value of about 500 lb/sq in. was available). Thus it appears that the water spray will strike the blade suction surface unless the spray is adequately accelerated by the gas stream. This figure will subsequently be referred to when the experimental results obtained from this investigation are discussed.

Experimental Results

The modified J33 engine under investigation was instrumented to provide blade temperature distributions of internally air-cooled blades as well as solid Haynes-Stellite 21 blades with spray cooling. Because of the large range of cooling and engine conditions considered and the great number of temperature distributions obtained, only those for the injection configuration giving the most favorable results over the entire speed range are presented. These data may be applicable to other turbine engines because of the wide range of engine speeds considered.

Air-cooled blades. - The effectiveness of water-spray cooling with internally air-cooled blades was investigated at engine speeds of 4000 and 8000 rpm (tip speeds of 454 and 908 ft/sec) as a means of temporarily augmenting the cooling achieved at the blade leading and trailing edges. Typical chordwise and spanwise temperature distributions of internally air-cooled blades obtained at these speeds with two inner-diaphragm injection nozzles (see fig. 3) are shown in figure 8. A constant air-coolant-to-gas flow ratio of 0.050 was provided, and blade temperature distributions are shown for water-coolant-to-gas flow ratios of zero and 0.0071. The midchord blade temperature on the blade pressure surface is plotted against the blade span in figure 8(a). The midchord position was selected as representing the temperature of the critical portion of the blade. Space limitations imposed by the rotating thermocouple pickup prevented obtaining a similar spanwise temperature distribution for the midchord position on the blade suction surface. For the spray-cooled condition at both 4000 and 8000 rpm, the maximum temperature difference in the blade is approximately 350° F. A temperature difference of 200° F occurs at the same speeds for the simple air-cooled case. The greatest degree of cooling is achieved over the lower portion of the blade, which is nearer the spray source. Some cooling is achieved near the blade tip as the water spray is swept outward along the blade surface by centrifugal force. Similar trends of greater cooling over the lower portion of the blade occur with the other injection configurations investigated, and there is little difference between injection configurations insofar as the radial temperature distribution of air-cooled blades is concerned.

The chordwise temperature distributions around the blade midspan shown in figure 8(b) are representative of the trends occurring at all water-coolant-to-gas flow ratios investigated and at all spanwise positions with this injection configuration. Cooling is achieved to a maximum extent near the leading edge on the suction surface with spray cooling. Decreases in blade temperature of about 730° and 520° F from the simple air-cooled case occur at this point during operation at 4000 and 8000 rpm, respectively. The blade trailing-edge temperature is reduced from the simple air-cooled case about 200° F on the pressure surface and about 120° F on the suction surface. As shown in figure 7, which is calculated on the basis that the water spray assumes the direction of the gas flow, reasonably low water velocities were required at these speeds to provide good coolant coverage on the blade pressure surface. The chordwise temperature distribution in figure 8(b), however, indicates that the water spray velocity is not sufficiently high for the water to cool effectively on the pressure surface. It should also be noted that the air-cooled-blade leading edge is more blunt than that of a solid blade. Also, when such a blade is placed between two solid blades, the passage configuration is somewhat altered. These factors probably affect the water-coolant coverage of the air-cooled blades as well. At both speeds investigated the suction surface of the internally air-cooled blades is cooled more noticeably with spray cooling than the pressure surface. It should be noted that conduction from one surface of this type of blade to the other is prevented by the air space. Slightly less effective cooling was obtained with the other injection configurations investigated, but no significant general change in the cooling pattern was apparent.

No temperature data are available for spray cooling of air-cooled blades at rated speed. Attempts at rated-speed (11,500 rpm) operation resulted in repeated blade failures. Figure 9 shows cracks at the root of an air-cooled blade which were characteristic of all the air-cooled-blade failures. The failures occurred in all cases at the brazed fillet between the shell and base of the blade and appear to have originated near the leading edge. This is probably due to unequal thermal expansion resulting from the high temperature differences occurring at this point on the blade, as indicated by the chordwise temperature distribution of figure 8(b). As a consequence of these repeated failures, it does not appear that water-spray cooling utilizing injection configurations such as those employed in this investigation is a satisfactory method of augmenting the cooling at the leading and trailing edges of this type of internally air-cooled blade.

Standard blade spray pattern. - Use of ordinary water in the spray-cooling process results in the deposition of minerals on the blade surface. The pattern obtained is herein referred to as the blade spray pattern and provides a visual means of interpreting the blade temperature distributions obtained. Figure 10 shows such a spray pattern obtained at rated-speed conditions with a combination of trailing-edge and inner-diaphragm injection configurations. This particular pattern was chosen for illustration because it provided the greatest clarity of all those available. The spray pattern on the suction surface is shown in figure 10(a). An area of deposition is apparent over the blade surface. Each heavy white line consists of mineral deposits from the water and probably represents the boundary between the wetted and unwetted portions of the blade surface. The orderly pattern of the lines of deposition evidently correspond to different values of coolant flow. As the flow rate was increased, more of the blade surface was wetted by the coolant. After a certain rate was reached, further increases probably resulted in the spray being swept off the blade without wetting any more of the blade surface. Figure 10(b) shows the pattern on the pressure surface of the blade for the same operating condition. A definite boundary between the wetted and unwetted surfaces is not apparent from the spray pattern. The vague pattern indicates that some water apparently evaporated from this surface without actually wetting the surface. Thus it appears from a consideration of the spray pattern that there is a definite tendency for the water to strike the blade suction surface, indicating that the water does not enter the blades at the aerodynamic gas entrance angle.

Spray cooling of standard blades. - In figure 11 are presented the spanwise and chordwise blade temperature distributions of standard rotor blades obtained with the inner-diaphragm type of injection configuration at engine speeds of 4000, 8000, and 11,500 rpm. The data are presented for coolant-to-gas flow ratios of 0.0072, 0.0077, and 0.0089 for these respective engine speeds and for zero coolant flow at each of these speeds. An attempt was made to provide a comparison at a constant coolant-to-gas flow ratio for all speeds; however, the coolant-to-gas flow ratio of 0.0089 (2360 pounds of coolant per hour) was the minimum obtained at 11,500 rpm and 0.0077 was the maximum obtainable at 8000 rpm with this type of injection configuration. The spanwise temperature distribution is shown in figure 11(a). At the blade tip and root sections for the runs at 4000 and 8000 rpm, only one midchord temperature was available; consequently all temperatures plotted are for the midchord position on the suction surface. It appears that spray cooling is rather effective since a maximum temperature reduction of approximately 900° F is obtained at rated speed with a coolant-to-gas flow ratio of less than 1 percent. The smallest radial temperature difference (150° F) occurs at rated speed. This is only 50° F greater than the radial temperature difference occurring at the uncooled condition and should not have a noticeable effect upon the blade stress along the midchord position. A more pronounced radial temperature difference occurred at rated speed with the other type of injection configuration investigated. The appreciable temperature difference of 440° F which occurred at an engine speed of 8000 rpm could probably be decreased by operation at a somewhat higher coolant flow rate. A somewhat smaller temperature difference occurred with other injection configurations investigated at a similar coolant-to-gas flow ratio and an engine speed of 8000 rpm, although the temperature difference was more pronounced at rated speed with these configurations.

A chordwise temperature distribution around the blade midspan is shown in figure 11(b) for the spray-cooled and the uncooled case. With a coolant-to-gas flow ratio of 0.0089, the blade trailing-edge temperature is 850° F at rated speed and represents a reduction of 400° F in the trailing-edge temperature. A maximum temperature difference of about 600° F between the leading and trailing edge and an average midspan blade temperature of 500° F, as compared with an uncooled blade temperature of about 1260° F, occurred, which makes the inner-diaphragm injection configuration appear relatively promising. Runs with incomplete instrumentation indicated that increased coolant flow decreased the leading-edge temperature but did not reduce the trailing-edge temperature appreciably, thereby increasing the temperature difference across the blade. The temperature distributions on the pressure and suction surfaces of standard blades when spray cooled are generally more symmetrical about the leading edge than for the case of internally air-cooled blades which are spray cooled. This is due to conduction through the blade metal which prevents large differences in temperature level between the blade surfaces.

Another feature shown here is the effective cooling achieved at the blade trailing edge at 4000 and 8000 rpm, and the much less effective cooling of the trailing edge which occurs at 11,500 rpm. The changing shape of the chordwise-temperature-distribution curve may be explained by a consideration of figure 7 and conjecture as to the probable path of the water spray as well as reference to the spray pattern shown in figure 10. Figure 7 indicates that for engine speeds of 4000, 8000, and 11,500 rpm, water velocities of 700, 1400, and 2000 feet per second, respectively, are required for the water to strike the blade near the aerodynamic gas entrance angle. It is assumed that, if the water velocity is sufficient to strike the blade at this angle (123° at the midspan blade position), it will be deposited along the pressure surface of the blade and will effectively cool the trailing edge. If the water spray strikes the rotor blade at a smaller angle, however, so that it is deposited along the suction surface, it is less likely to follow the blade contour and may be swept off the blade near the midchord position. As a result, the trailing edge will not be well cooled. This conjecture seems to be corroborated by the typical spray pattern obtained with a standard blade at rated speed. The velocity of 700 feet per second required for water entrance at the aerodynamic gas entrance angle at 4000 rpm is probably attained by the water spray because of the acceleration by the gas stream. At 8000 rpm the trailing edge is not so well cooled and the suction-surface midchord temperature is slightly lower than the pressure-surface midchord temperature. In this case it is probable that some of the spray is accelerated to a speed of 1400 feet per second, thus wetting the pressure surface, while the remainder is not sufficiently accelerated, causing it to strike the suction surface. At 11,500 rpm the required water velocity of 2000 feet per second apparently is not achieved, and only the leading portion of the rotor blade (particularly on the suction surface) is effectively cooled, as indicated by the spray pattern of figure 10.

The inner-diaphragm type of injection configuration was the only one which cooled the trailing edge effectively at both 4000 and 8000 rpm. The large temperature difference across the blade as a result of the high leading-edge temperatures at these speeds can probably be decreased by use of a combination of stator trailing-edge and inner-diaphragm injection nozzles since the former type always provided excellent cooling at the rotor-blade leading edge when tested individually. Such a nozzle combination was not checked at low engine speeds. At rated speed, however, this suggested nozzle configuration provided lower leading-edge temperatures (by about 100° F) than the simple inner-diaphragm configuration. The disadvantage of installation difficulties with stator trailing-edge nozzles, however, overbalances the slight gain in cooling obtained. In view of the high chordwise temperature gradients obtained at rated speed with all the injection configurations investigated, further research is indicated.

Two blade failures, one an instrumented blade with considerable blade metal removed to accommodate the thermocouple installation, occurred during rated-speed operation, probably as a result of large blade temperature differences, with an injection configuration consisting of stator-blade suction-surface nozzles. Whether or not similar failures may occur with the inner-diaphragm type of injection configuration under more severe operating conditions can only be determined by actual test. It should be noted that under certain conditions, combat maneuvers, for example, more severe shock conditions will be encountered by the blades in an aircraft spray-cooling application than in this investigation. This is due to the necessity of injecting the water spray while the turbine is being operated at maximum power and the blades are approximately at gas temperature. Failure due to thermal shock is recognized as a separate problem requiring investigation, but one that is not believed to be insurmountable. In the present investigation, however, an attempt was made to minimize this problem by injecting the water simultaneously with the engine starts in order that data as to the general cooling effectiveness of spray cooling might be obtained more readily.

Comparison of water-injection configurations. - A comparison of the reductions in blade temperature obtainable by water-spray cooling with several configurations at rated speed is given in figure 12. The difference between the effective gas temperature $T_{g,e}$ and the average blade temperature $T_{B,av}$ integrated around the blade periphery at the midspan position is plotted against the coolant-to-gas flow ratio. The data are plotted as a temperature difference to mask out the effects of small changes in gas temperature which occurred during engine operation with these injection configurations. The comparison is valid only for the conditions at which the engine was operated and does not reflect the existence of large temperature differences in the blade. It cannot be used as a correlation method suitable for extrapolation to other operating conditions.

For all the nozzle configurations considered, a rapid increase in blade cooling occurs up to a coolant-to-gas flow ratio of 0.010. Beyond this dilution the degree of cooling does not increase rapidly with increases in coolant-to-gas flow ratio. Of the injection configurations for which there were sufficient data to make this comparison, the one which employs inner-diaphragm nozzles indicates a high degree of effectiveness, as expected. However, it appears that the stator-trailing-edge configuration by itself is also highly effective. This results from the fact that the stator-trailing-edge configuration tends to cool the leading-edge portion of the rotor blade excessively, thus providing a lower average rotor-blade temperature and resulting in a large chordwise thermal gradient. Use of the inner-diaphragm type of injection nozzle provides more even cooling around the blade periphery.

The combination of stator suction-surface and trailing-edge nozzles provided the least effective cooling of all the configurations shown in the figure. The limited data (not shown) obtained with stator suction-surface nozzles alone at rated speed indicated slightly less effective cooling than when these nozzles were operated in combination with stator trailing-edge nozzles. Although a data comparison is not shown, it should be noted that the variation in the spanwise positioning of the nozzles along the stator-blade suction surface and trailing edge (three nozzles per blade spaced equal distances apart, but located closer to the blade base in one case than the other) resulted in slight differences in the cooling effectiveness achieved, the lower position being better in all cases. The limited data available indicate that the inner-diaphragm nozzles by themselves cooled slightly better than the combination stator trailing-edge and inner-diaphragm injection configuration. Neither the stator suction-surface nozzles nor the inner-diaphragm nozzles were operated over a complete coolant-to-gas flow range individually because of thermocouple instrumentation difficulties. Consideration of all the factors involved in a comparison of injection configurations indicates that the inner-diaphragm injection configuration besides cooling effectively is also the most desirable of the types investigated from the standpoint of installation and maintenance ease. Much of the complicated drilling and welding required for the other configurations is eliminated in the original installation. Also, difficulties with the development of welding cracks during engine operation such as occur around the stator trailing-edge nozzles are eliminated and almost unlimited operation without nozzle repair can be achieved.

Comparison of the manner in which the number of injection stations around the stator-ring circumference affects cooling cannot be shown readily by means of a plot similar to that of figure 12 because insufficient data are available. Results at 4000 and 8000 rpm indicated that slightly better cooling occurred when two rather than four injection stations around the stator-ring circumference were employed. Use of one injection station decreased the cooling effectiveness, indicating that concentration of the spray is beneficial only to a point. There is not sufficient data, however, to provide a complete comparison, and definite conclusions cannot be drawn.

Extrapolation of spray-cooled data. - In order that some indication may be obtained as to the amount of water required for engine operation in the range of high gas temperatures and overspeed, an extrapolation was made from data obtained at rated speed and at three gas temperatures with an injection configuration consisting of a combination of inner-diaphragm and stator trailing-edge nozzles. Figure 13 shows the experimental data upon which the extrapolation was based. The efficiency of the spray-cooled process

$$\sigma = 1.024\times10^{-6}\left(\frac{T_{g,e}^{0.75}}{T_B^{0.425}}\right)\left(\frac{T_{g,e} - T_B}{\dfrac{w_w}{w_g^{0.75}}}\right)$$

is plotted against $w_w/w_g^{0.75}$ for several effective gas temperatures. The efficiency reaches a maximum for a $T_{g,e}$ of 1260° F at a value of the coolant-to-gas flow parameter of 0.0275. More cooling can be effected at higher values of the coolant-to-gas flow parameter, although the efficiency of the process decreases. Additional data to verify this trend are not available since the rated-speed data were obtained last and failure of the weld seams around the stator-trailing-edge-nozzle injection stations prevented further operation.

The results of the data extrapolation with the lowest or most conservative curve of figure 13 are presented in figure 14. The required coolant-to-gas flow ratio is plotted against turbine inlet-gas temperature for various overspeed conditions. A coolant-to-gas flow ratio of 3.15 percent (equivalent to a coolant flow rate of 20 gal/min) is required for operation at an overspeed of 10 percent and a turbine inlet-gas temperature of 2000° F. This coolant flow is approximately 30 percent larger than the calculated fuel flow at this operating condition. Engine operation at 5 percent overspeed and at the same gas temperature can be achieved with less than one half the amount of coolant flow required for the condition of 10 percent overspeed.

The positioning of these curves is largely dependent upon the value of the stress-ratio factor selected since it is necessary for determining the allowable blade stress. Since the extrapolation method is based on an average midspan blade temperature, a stress-ratio factor of 3.25 was arbitrarily selected in order to account for additional blade stresses induced by temperature gradients such as existed in the blades at rated engine speed. This value is more than twice as great as that employed for an air-cooled-blade design which has been operated at rated engine conditions (ref. 5) and about three times as great as that employed for standard Haynes-Stellite 21 blades in the J33 engine. Whether use of such a safety factor is adequate to account for the adverse effects of high temperature differences is not known, and therefore too much significance should not be placed upon the rather optimistic results indicated in figure 14. However, these results serve to indicate that relatively small quantities of water are involved in the application of spray cooling to achieve the large thrust increase available from engine operation at conditions above rated. To achieve an inlet-gas temperature of 2000° F and an overspeed condition of 10 percent with the injection configuration investigated requires a spray-cooling-process efficiency of approximately 13 percent, if a stress-ratio factor of 3.25 is assumed. This value is considerably less than the maximum efficiency attainable shown in figure 13. It is interesting to note what spray-cooling efficiency may be required at the maximum engine operating condition for rated speed calculated in figure 5(a) (inlet-gas temperature of 2400° F) with the limiting 3-percent dilution designated. If a more realistic stress-ratio factor of 2 is adequate to account for the effect of stresses introduced by temperature gradients, a spray-cooling-process efficiency of approximately 15 percent would be required at this operating condition.

In general, the experimental phase of the investigation indicated that the best cooling from an over-all standpoint was achieved with the inner-diaphragm type of injection configuration. Even with this configuration a high stress-ratio factor will probably be required to account for large temperature differences in the blade if operation at overspeed, over-temperature conditions is contemplated. Finally it appears that coolant-to-gas flow requirements are of the same order of magnitude as those originally deemed realistic and considered in the analytical portion of the investigation.

SUMMARY OF RESULTS

The following results were obtained from an investigation of water-spray cooling conducted with a modified J33-A-9 turbojet engine:

1. With the assumption of adequate spray cooling at a coolant-to-gas flow ratio of 3 percent, calculation for a modified J33 engine at sea-level static conditions indicated that a thrust increase up to 40 percent over rated thrust may be achieved by engine operation at an inlet-gas temperature of 2000° F and an overspeed of 10 percent. A calculated thrust increase of 52 percent may be achieved with a total coolant-to-gas flow ratio of 7 percent if compressor water injection can also be applied at these conditions.

2. Water-spray cooling as a means of temporarily augmenting cooling at the leading and trailing edges of air-cooled blades was not satisfactory for the water-injection configurations used in this investigation. Repeated blade failures occurred at the brazed junction between the blade shell and base, probably as a result of large blade temperature differences when rated-speed operation was attempted with spray cooling. Two standard Haynes-Stellite 21 blades (one an instrumented blade having considerable metal removed) failed during spray-cooled operation at rated engine speed. Both failures were probably caused by large temperature differences induced by spray cooling.

3. With the use of inner-diaphragm nozzles, water-spray cooling of standard Haynes-Stellite 21 blades resulted in an average midspan blade temperature of about 500° F at rated-speed operation with a coolant-to-gas flow ratio of 0.0089, as compared with an uncooled blade temperature of about 1260° F. A maximum temperature difference of about 600° F between the blade leading and trailing edge occurred at this operating condition. No appreciable radial temperature difference was observed at the rated condition.

4. All injection configurations investigated provided high chordwise temperature gradients in the blades at rated-engine-speed operation.

5. An injection configuration consisting of spray nozzles located in the inner ring of the stator diaphragm appeared to be the best of all the configurations investigated from the over-all standpoint of cooling effectiveness, installation, and maintenance.

6. Calculations based on strength properties of the blade material, an arbitrary stress-ratio factor, and extrapolation of rated-speed spray-cooling data indicated a spray-cooling rate of 20 gallons per minute was required to permit operation of this engine at an inlet-gas temperature of 2000° F and an overspeed condition of 10 percent. This flow rate is approximately 30 percent larger than the calculated fuel-flow requirements at this operating condition.

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APPENDIX - SYMBOLS

A	cross-sectional area, sq ft
a	velocity of sound, ft/sec
c_p	specific heat at constant pressure, Btu/(lb)(^{o}F)
c_v	specific heat at constant volume, Btu/(lb)(^{o}F)
F	thrust, lb
$\overline{F}$	constant
f/a	fuel-air ratio
g	acceleration due to gravity, ft/sec^2
H	gas-to-blade convection heat-transfer coefficient, Btu/(sq ft)(sec)(^{o}F)
h'	enthalpy based on stagnation condition, Btu/lb
h_{fg}	heat of evaporation, Btu/lb
J	mechanical equivalent of heat, 778 ft-lb/Btu
K_1 to K_4	constants
k	thermal conductivity, Btu/(ft)(sec)(^{o}F)
M	Mach number, V/a
N	engine speed, rpm
Nu	Nusselt number, Hx/k
Pr	Prandtl number, $c_p\mu/k$
p	gas static pressure, lb/sq ft
p'	gas total pressure, lb/sq ft
Q	rate of heat flow, Btu/hr
R	gas constant, ft/^{o}R
Re	Reynolds number, $\rho Vx/\mu$

r	radius, ft
S	blade surface area, sq ft
T	temperature, or gas static temperature, $^{\circ}R$ or $^{\circ}F$
T'	gas total temperature, $^{\circ}R$ or $^{\circ}F$
T''	gas total temperature relative to rotor, $^{\circ}R$ or $^{\circ}F$
V	absolute velocity, ft/sec
W	relative velocity, ft/sec
w	weight flow, lb/sec
x	characteristic dimension, ft
Z	exponent
α	stator outlet angle, deg
β	angle relative to plane of rotor rotation, deg
γ	ratio of specific heats, c_p/c_v
δ	p'/p_0
η	efficiency
θ	T'/T_0
Λ	recovery factor
μ	viscosity, lb/ft-sec
ρ	gas density, lb/cu ft
σ	efficiency of spray-cooling process

Subscripts:

a	air
ac	actual
av	average
B	blade

c	compressor
e	effective
f	fuel
g	gas
r	rotor
s	isentropic
T	total
t	turbine
w	water
0	NACA sea-level air
1	compressor inlet
1a	at end of saturated compression
2	compressor outlet
3	turbine inlet
4	turbine outlet
5	at exhaust nozzle

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TABLE I. - RANGE OF OPERATING CONDITIONS

Blade type	Engine speed, rpm	Water-injection configuration[a]	Effective gas temperature, °F	Air-coolant-to-gas flow ratio	Water-coolant-to-gas flow ratio
Air cooled (fig. 2(a)) and standard (fig. 2(b))	4000	One T.E. (low)	1130	0.05	0.0017 to 0.0056
		Two T.E. (low)	1150		.0031 to .011
		Four T.E. (low)	1150		.0044 to .017
		One S.S. (low)	1125		.0010 to .0043
		Two S.S. (low)	1150		.0023 to .0085
		Two S.S. (low) and two S.S. (high)	1150		.0030 to .017
		One I.D.	1140		.0021 to .0084
		Two I.D.	1150		.0047 to .017
	8000	Two T.E. (low)	1040		0.0021 to 0.0062
		Four T.E. (low)	1090		.0015 to .012
		Two T.E. (low) and two T.E. (high)	1020		.0027 to .010
		Two S.S. (low) and two S.S. (high)	1070		.0032 to .0089
		Two I.D.	1050		.0019 to .0077
Standard (fig. 2(c))	11,500	Two T.E. (low)	1240 to 1270		0.0071 to 0.0109
		Four T.E. (low)	1270 to 1470		.0012 to .0185
		Two T.E. (low) and two S.S. (low)	1220		.0059 to .0217
		Two T.E. (low) and two I.D.	1260 to 1470		.0053 to .0214
		Two I.D.	1272		.0089

[a]T.E., stator-blade trailing-edge nozzles; S.S., stator-blade suction-surface nozzles; I.D., nozzles located in inner ring of stator diaphragm; (low), nozzle group located nearer to blade base; (high), nozzle group located nearer to blade tip.

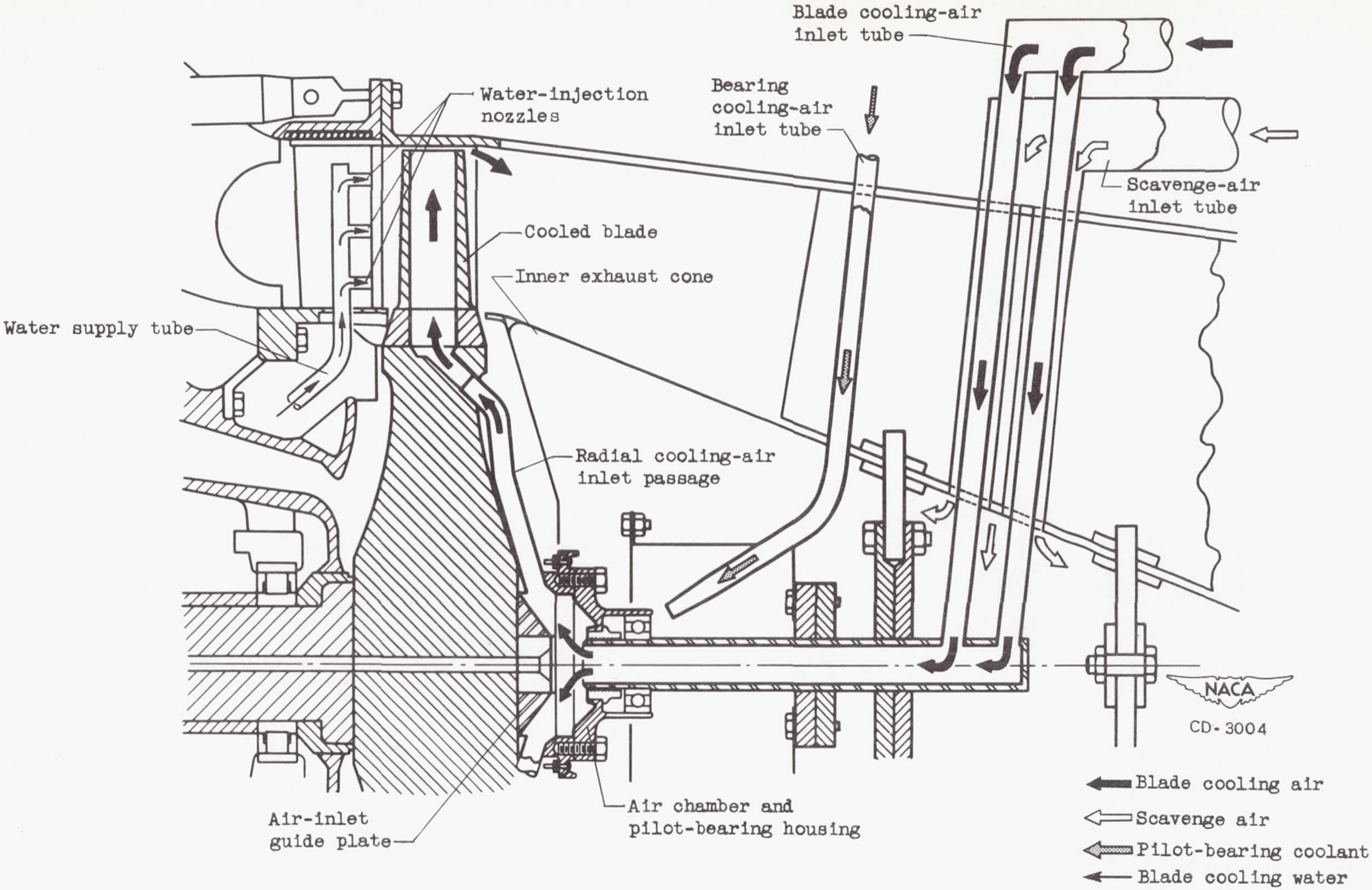
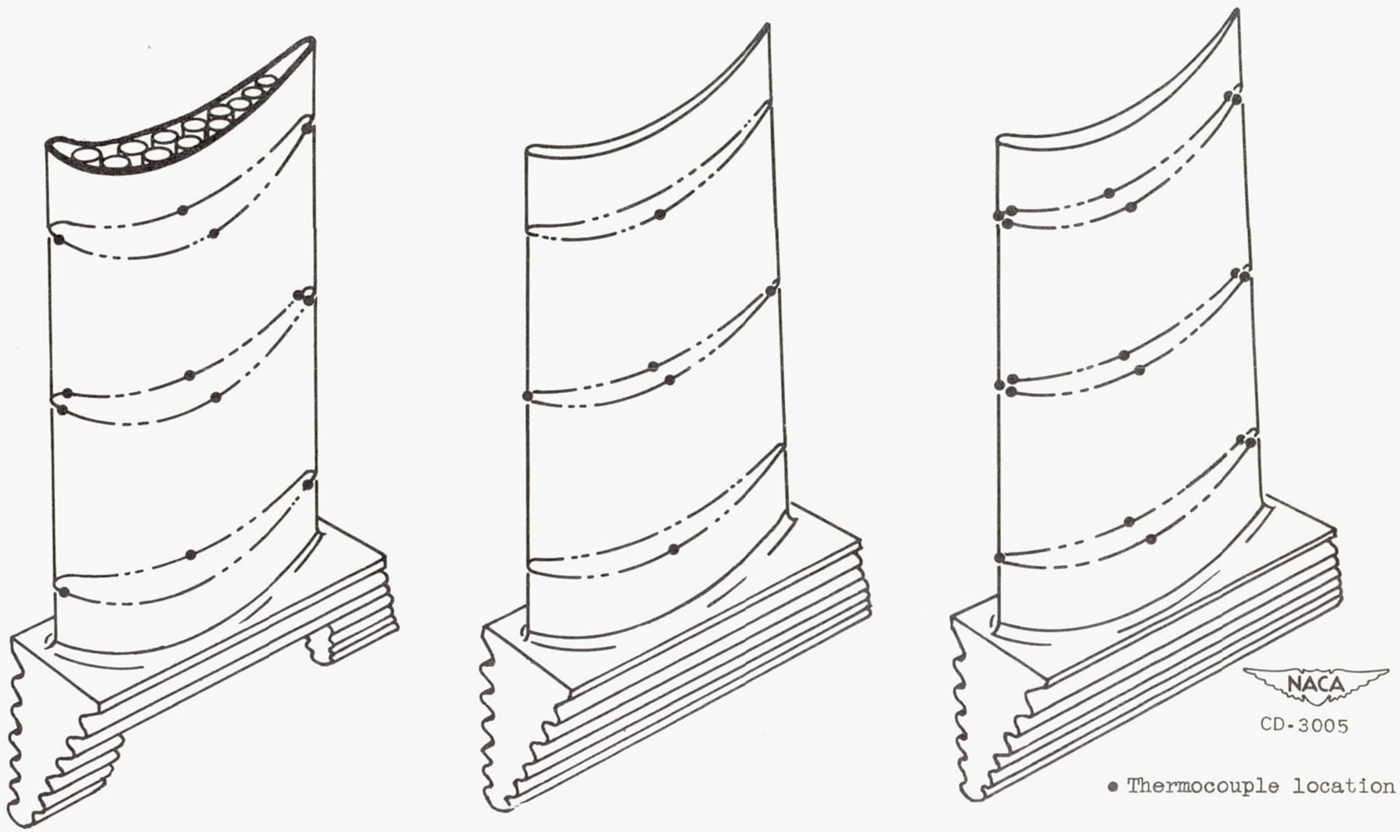


Figure 1. - Section of J33-A-9 engine showing a water-spray-injection configuration and modifications necessary to provide cooling air to four blades.



(a) Air-cooled blade with thermocouple-instrumentation locations used at engine speeds of 4000 and 8000 rpm.

(b) Solid blade with thermocouple-instrumentation locations used at engine speeds of 4000 and 8000 rpm.

(c) Solid blade with thermocouple-instrumentation locations used at engine speed of 11,500 rpm.

Figure 2. - Types of rotor blade used in investigation.

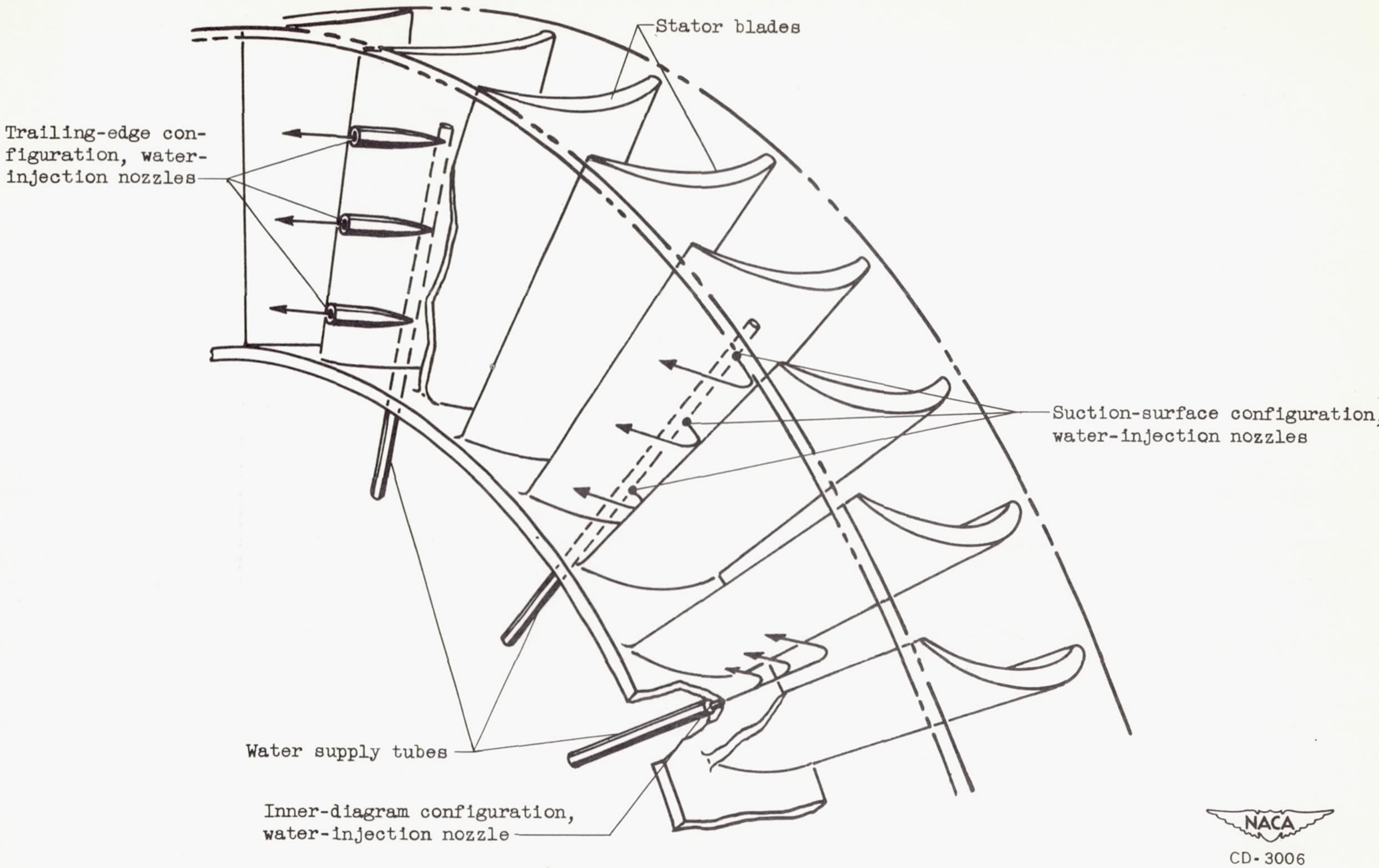


Figure 3. - View of the types of water-spray-injection configuration investigated.

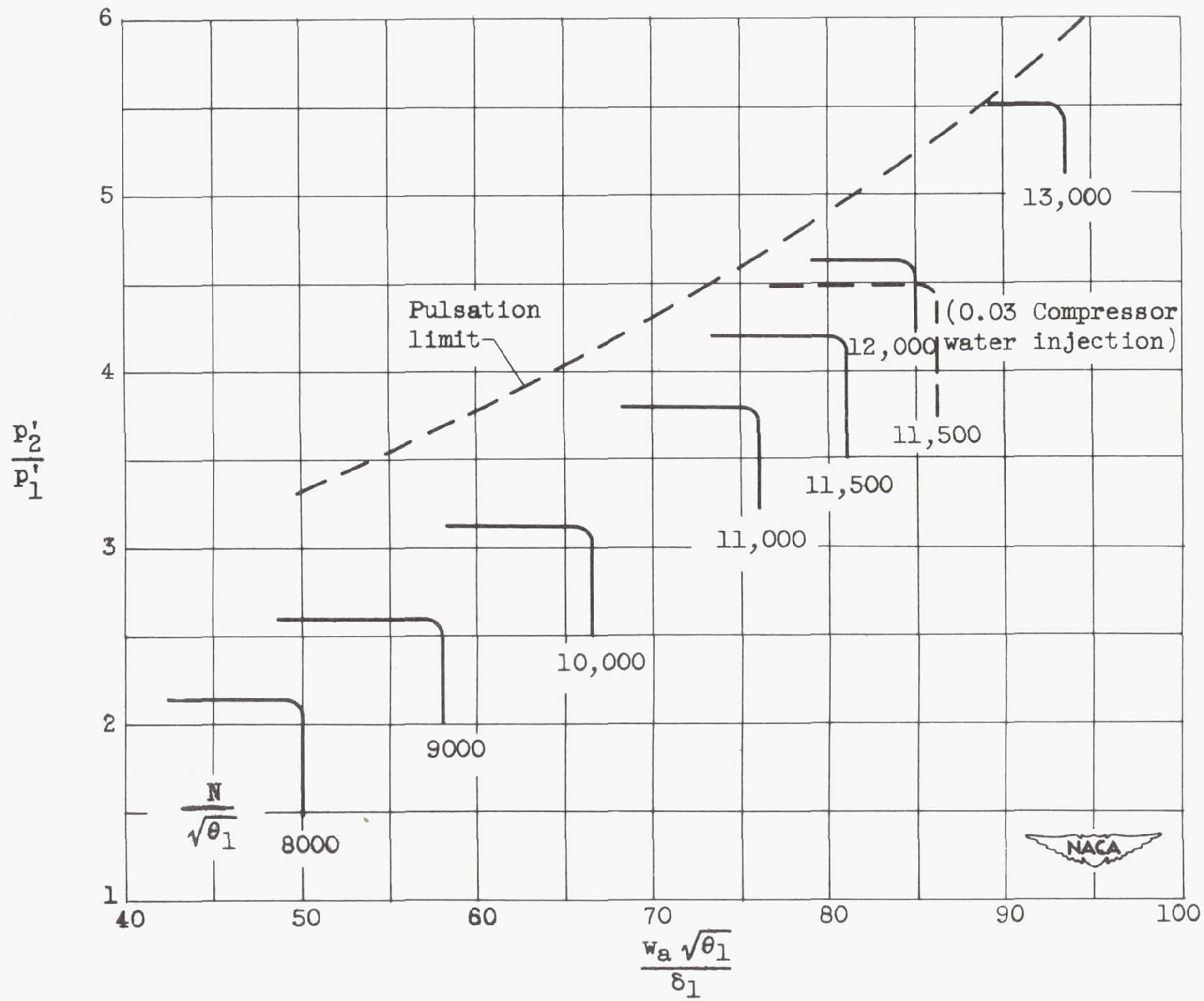
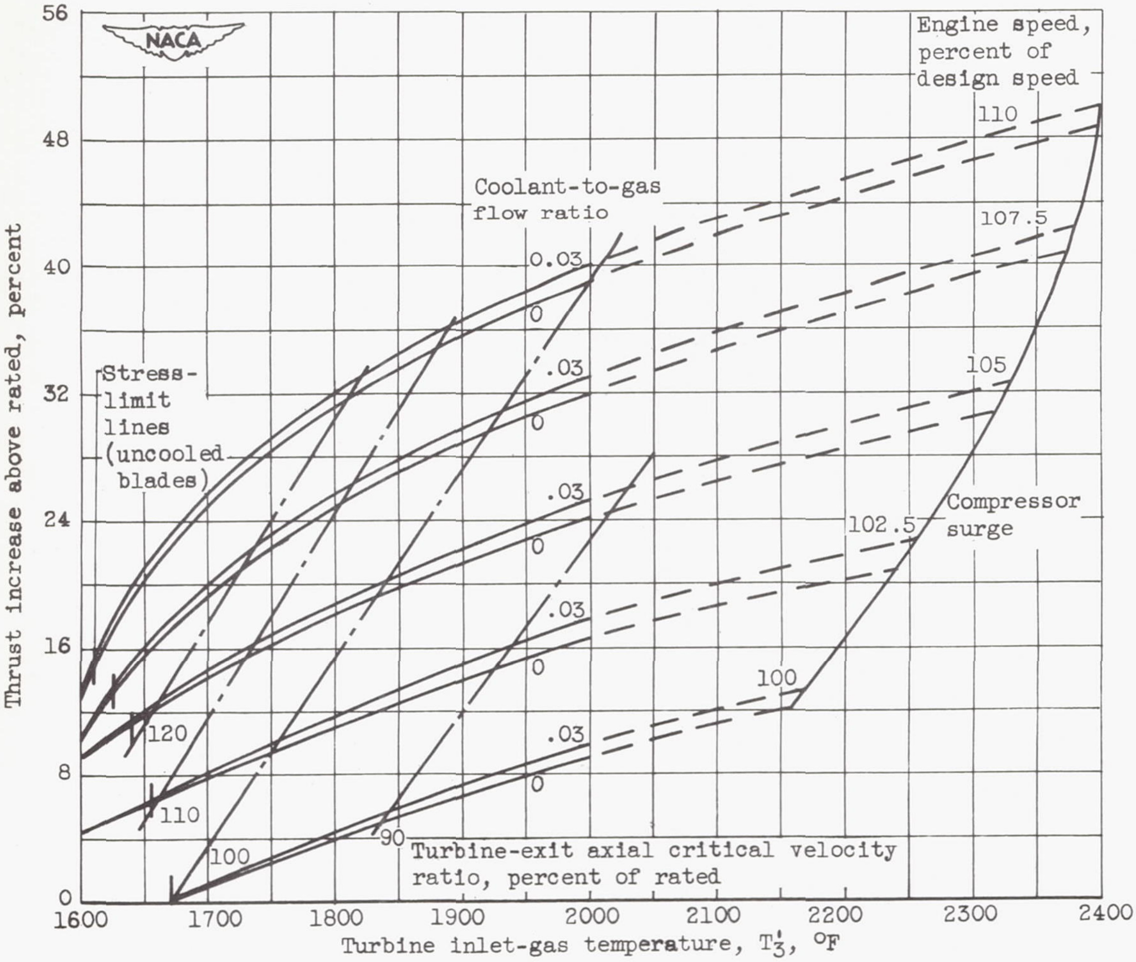
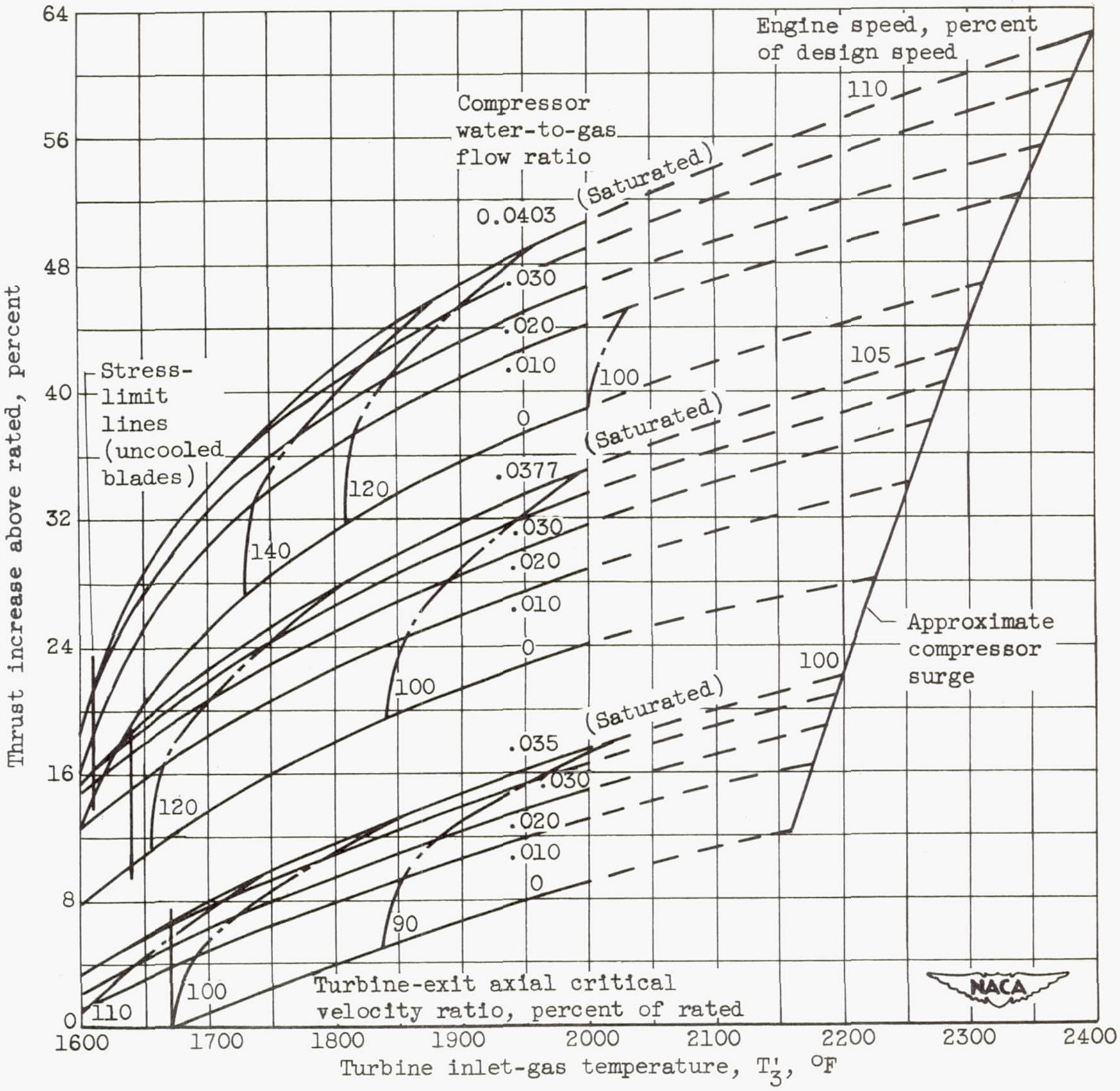


Figure 4. - Compressor characteristic lines (ref. 8, fig. 2, p. 16).



(a) Thrust increase available with and without spray cooling provided engine could be operated over this range of conditions.

Figure 5. - Calculated sea-level static-thrust increase available with J33-A-9 engine at elevated operating conditions.



(b) Thrust increase available with compressor water injection provided engine could be operated at these conditions without cooling.

Figure 5. - Concluded. Calculated sea-level static-thrust increase available with J33-A-9 engine at elevated operating conditions.

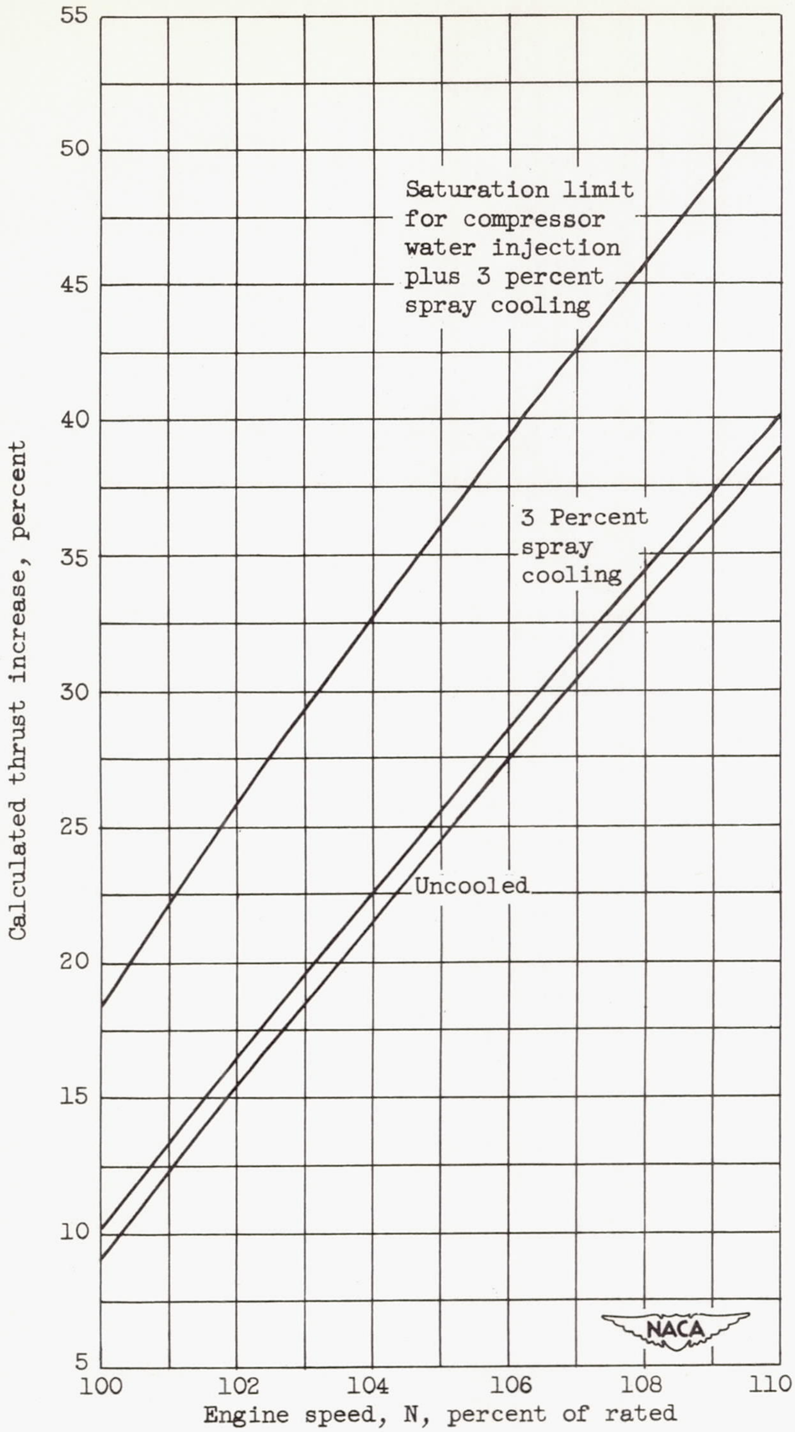


Figure 6. - Comparison of calculated static sea-level thrust increase available with J33-A-9 engine at turbine inlet-gas temperature of 2000° F for various overspeed conditions.

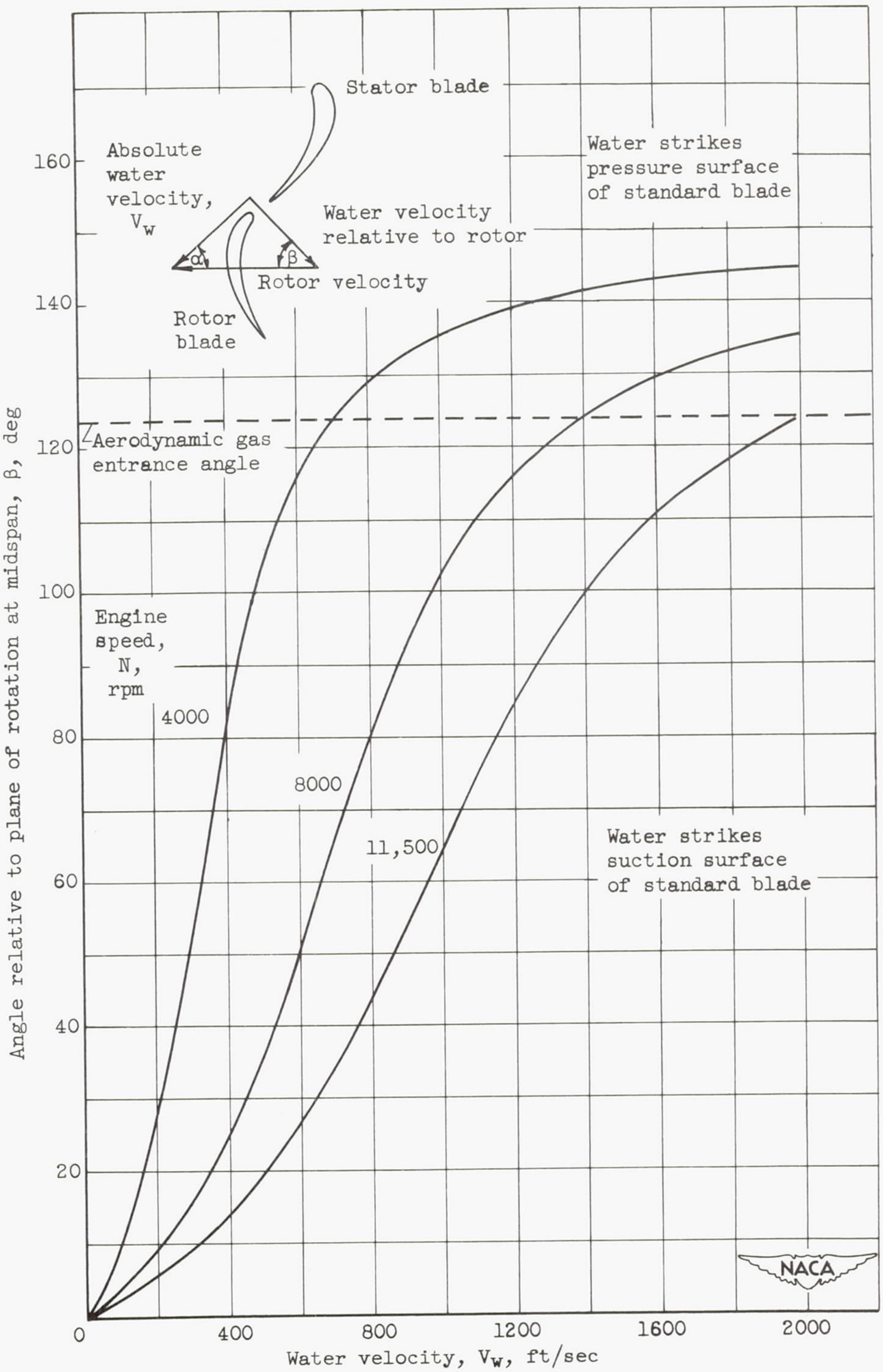
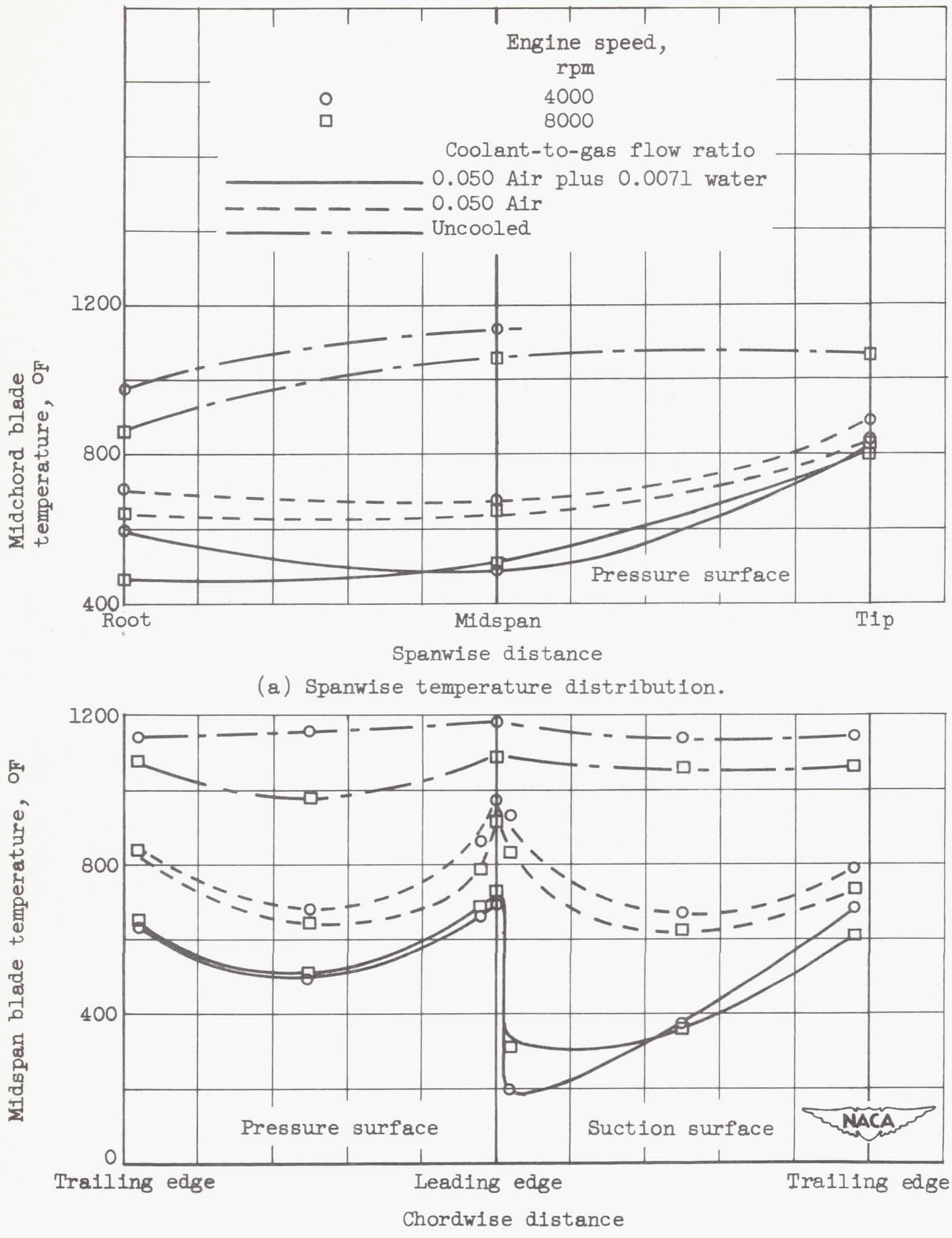


Figure 7. - Effect of water velocity on angle of incidence of water relative to plane of turbine rotation for engine speeds of 4000, 8000, and 11,500 rpm at the blade midspan position. Radius to midspan position, 0.9167 foot; absolute angle α, 29°.



(a) Spanwise temperature distribution.

(b) Chordwise temperature distribution.

Figure 8. - Typical temperature distributions for air-cooled blades obtained at engine speeds of 4000 and 8000 rpm with an air-coolant-to-gas flow ratio of 0.050 and a water-coolant-to-gas flow ratio of 0.0071. The inner-diaphragm type of water-injection configuration was used.

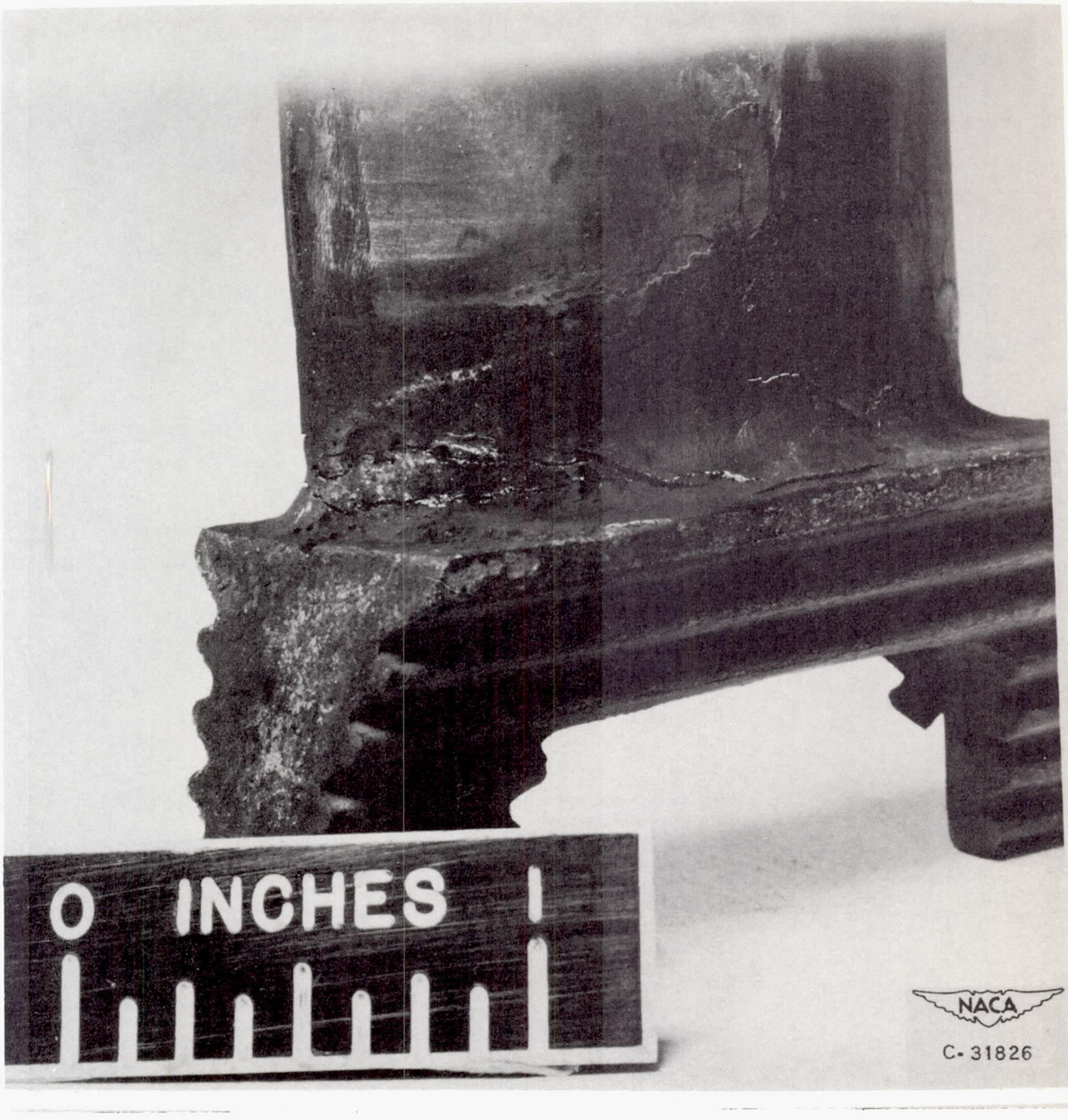
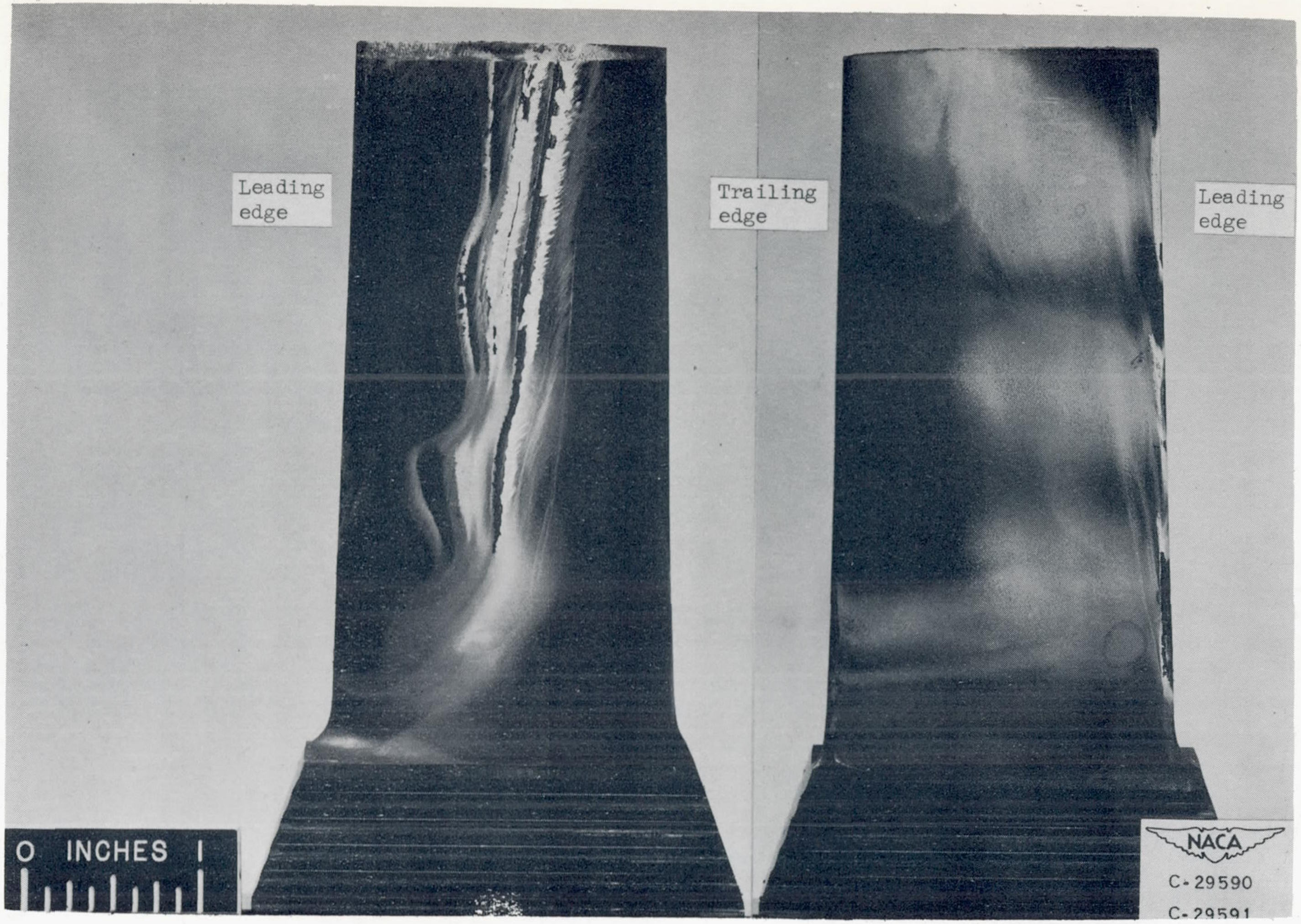
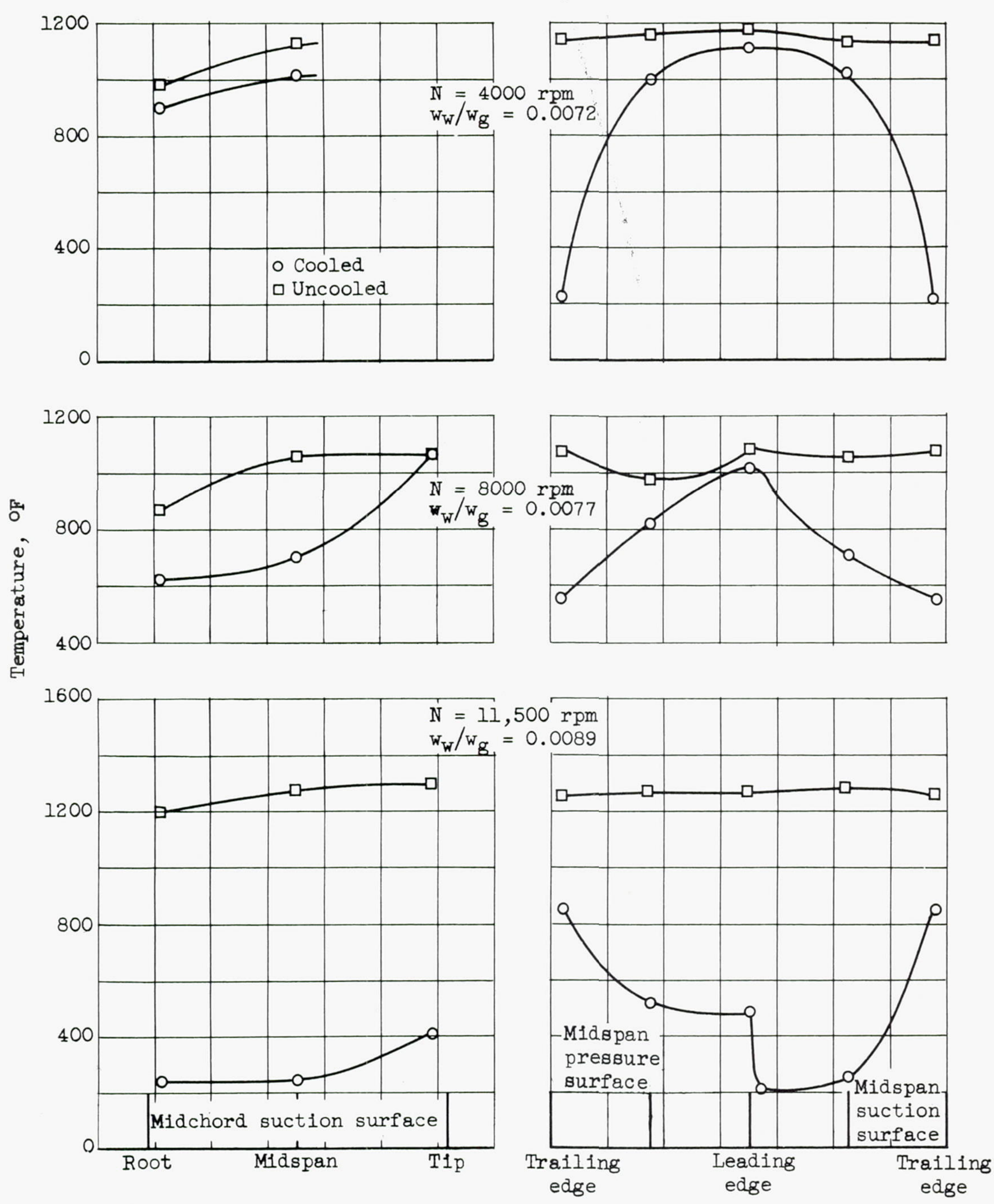


Figure 9. - Base of air-cooled blade showing cracks in fillet which are typical of air-cooled-blade failures.



(a) Suction surface. (b) Pressure surface.

Figure 10. - Deposits on surfaces of solid blade after operation at engine speed of 11,500 rpm with combination of trailing-edge and inner-diaphragm types of injection configuration.



(a) Spanwise blade temperature distribution.

(b) Chordwise blade temperature distribution.

Figure 11. - Typical temperature distributions for solid blades obtained at engine speeds of 4000, 8000, and 11,500 rpm. The inner-diaphragm type of injection configuration was employed. N, engine speed; w_w/w_g, coolant-to-gas flow ratio.

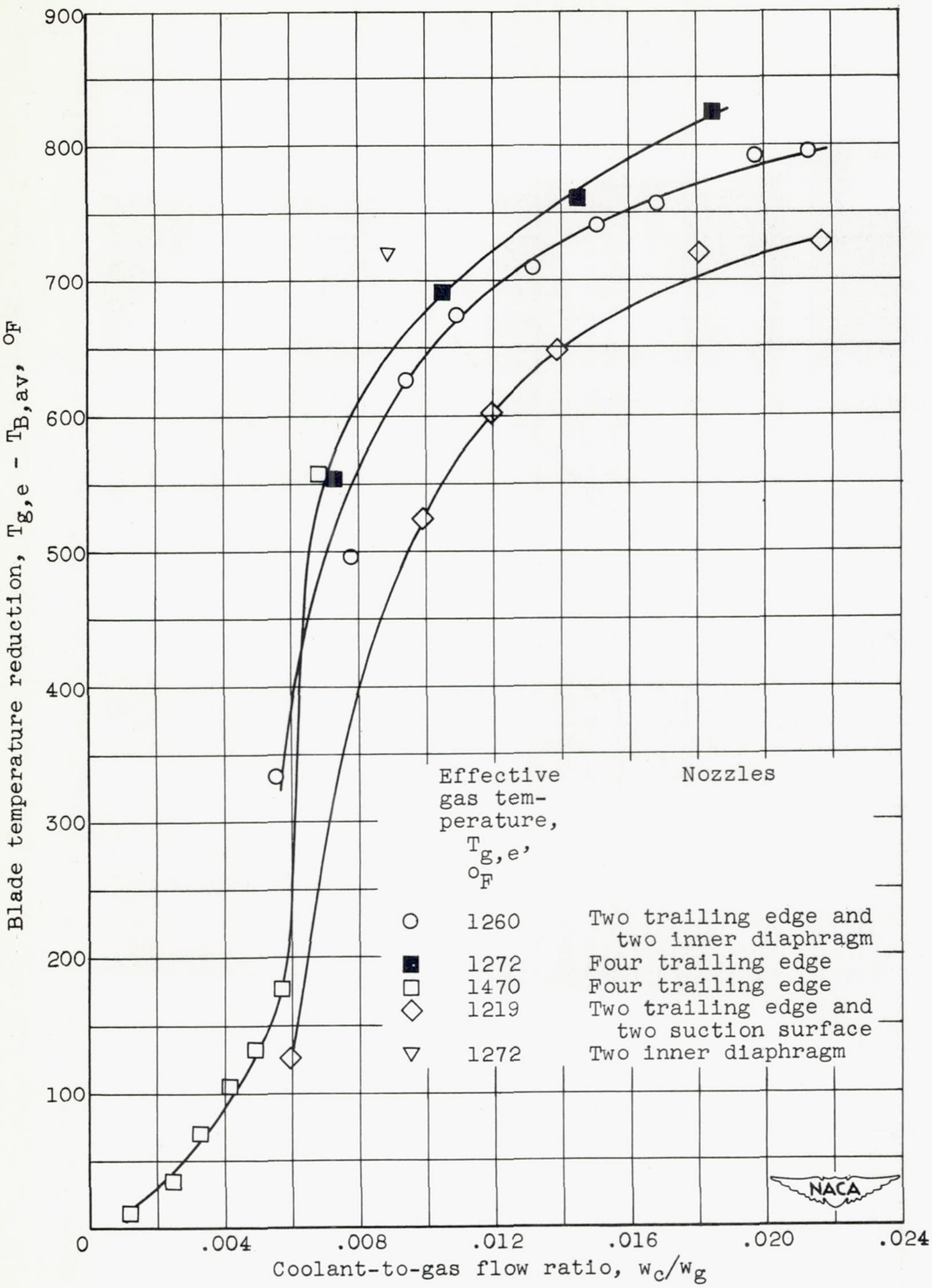


Figure 12. - Comparison of blade temperature reduction achieved with individual and combinations of water-spray-injection configurations at engine speed of 11,500 rpm at midspan position of solid blades.

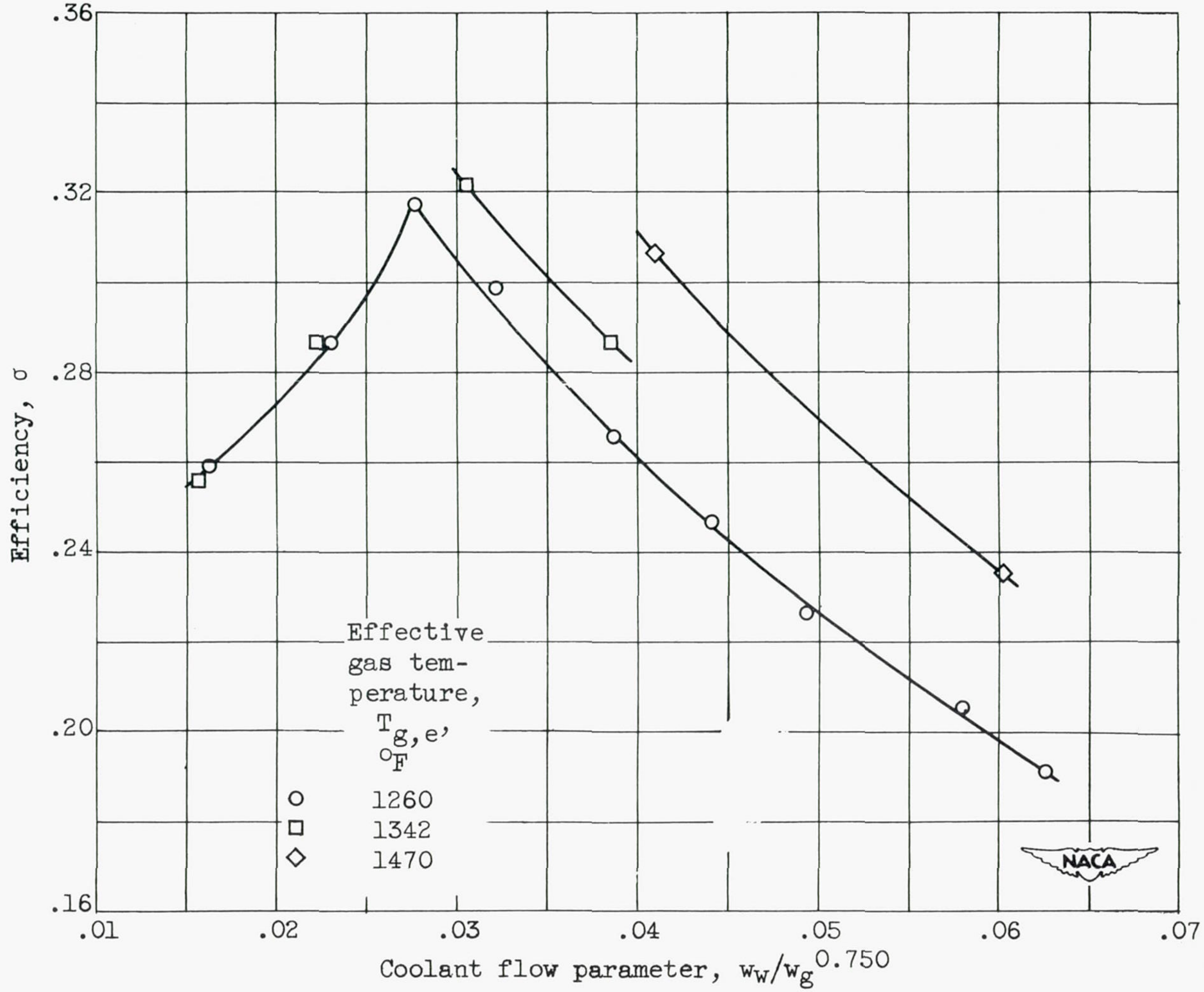


Figure 13. - Efficiency of spray-cooling process plotted against coolant flow parameter for combination of trailing-edge and inner-diaphragm types of injection configuration. Engine speed, 11,500 rpm.

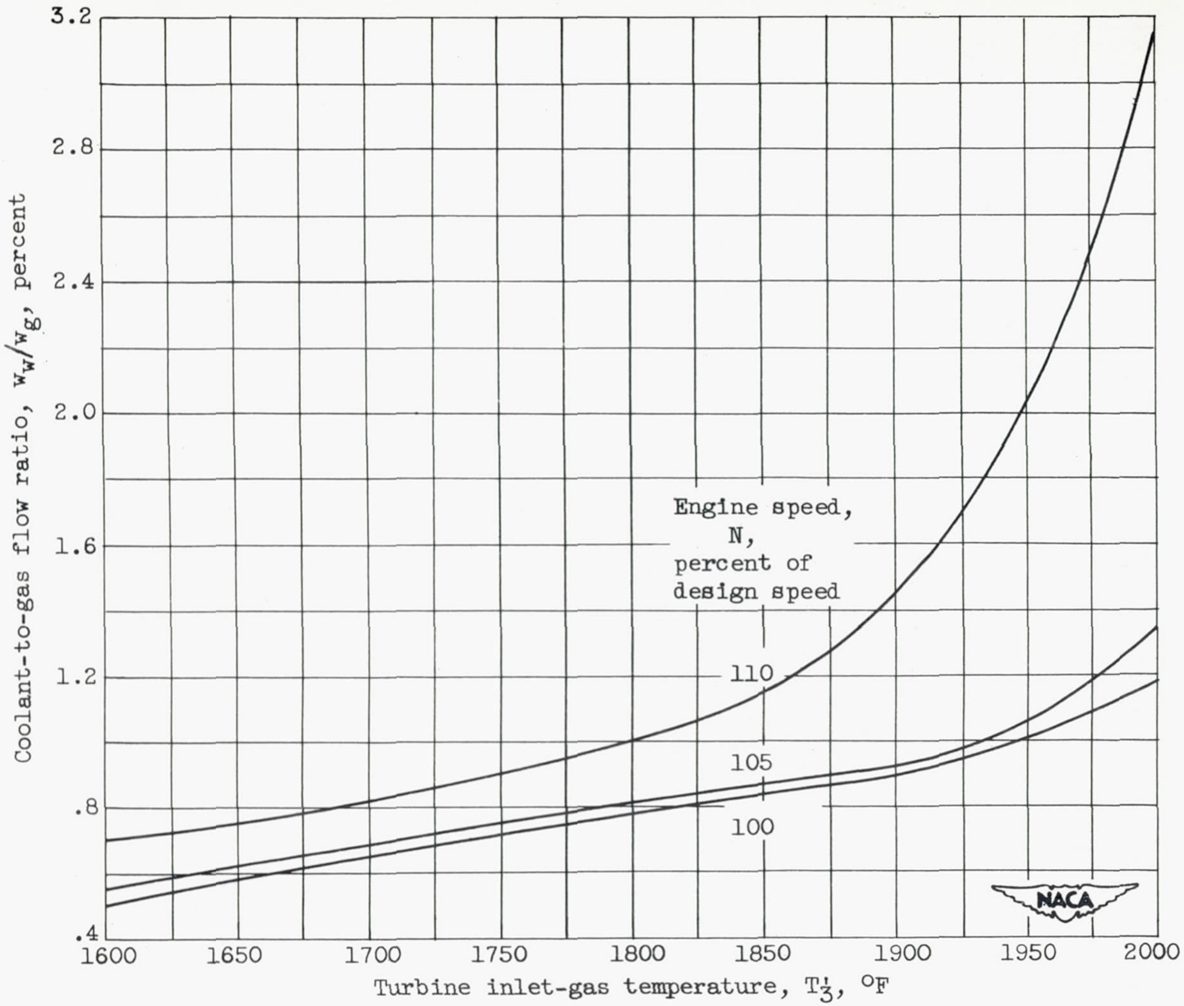


Figure 14. - Calculated coolant-to-gas flow ratios required for operation of J33-A-9 engine with spray cooling at conditions above rated. Stress-ratio factor, 3.25.